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TFE Sliding Surfaces in Bridge Bearings

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Abstract:	Tetrafluorethylene (TFE) is used widely to provide a sliding surface in a bridge bearing. This is due mainly to the material's low frictional property. This report presents the state-of-the-art of TFE sliding bearings. Parameters which may influence the coefficient of friction of TFE are discussed, and those parameters whose influences are not well documented in the literature are identified.
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LIST OF NOTATION

A	real area of the contact surface
A_0	geometrical area of the contact surface
F	frictional force
h	reduction in thickness
K	wear factor
k	parameter defined in Eqn.(3.1)
L	travel path
N	applied normal force
n	parameter defined in Eqns.(3.1) and (3.7)
P	average pressure at the sliding surface
p_y	compressive yield strength
Q	parameter ($= \pi k A_0$), defined in Eqn.(3.7)
R	surface roughness
S	stroke
T	temperature
t	time under loading
V	sliding velocity
W	wear rate ($= h/Vt$)
μ	coefficient of friction
τ	interface shear strength

1. INTRODUCTION

Bridges are not static structures. They experience movement due to thermal expansion and contraction, traffic induced deflection and oscillation, as well as the effects of prestressing, creep and shrinkage. Those movements are usually accommodated by the bearings supporting the bridge.

Bearings utilizing tetrafluoroethylene (TFE) to provide sliding surfaces are widely used in bridge structures. The merits of TFE as a material for a sliding surface in a bridge bearing are its low friction property, its chemical inertness under normal conditions, as well as its resistance to water absorption and weathering. TFE is incorporated in a variety of modern bearings, as shown in Figs.1 and 2. Figure 1 shows a pot and a spherical bearing, while Fig.2 shows a disc bearing and a slider bearing comprising a TFE sheet on a laminated elastomer bearing. The low friction TFE surface is used to provide either rotation by sliding over a spherical surface, or horizontal movement by sliding on a plane surface, or a combination of both. Stainless steel is commonly used for the surface mating with the TFE.

The Ontario Highway Bridge Design Code (OHBDC) [3] specifies design requirements for TFE sliding surfaces in bridge bearings. These requirements are based largely on current practices in Ontario [4] and Europe. Only limited data are available on the performance of TFE sliding surfaces in bearings constructed from North American materials and under service conditions experienced in Ontario. Consequently, the Ministry of Transportation and Communications (MTC) of Ontario has undertaken a research program to examine the state-of-the-art and refine the provisions in the OHBDC in relation to TFE sliding surfaces.

This report summarizes the work carried out at Queen's University on Phase I of a research project entitled "TFE Sliding Surfaces in Bridge Bearings" sponsored by the MTC. Phase I of the project comprised the following tasks:

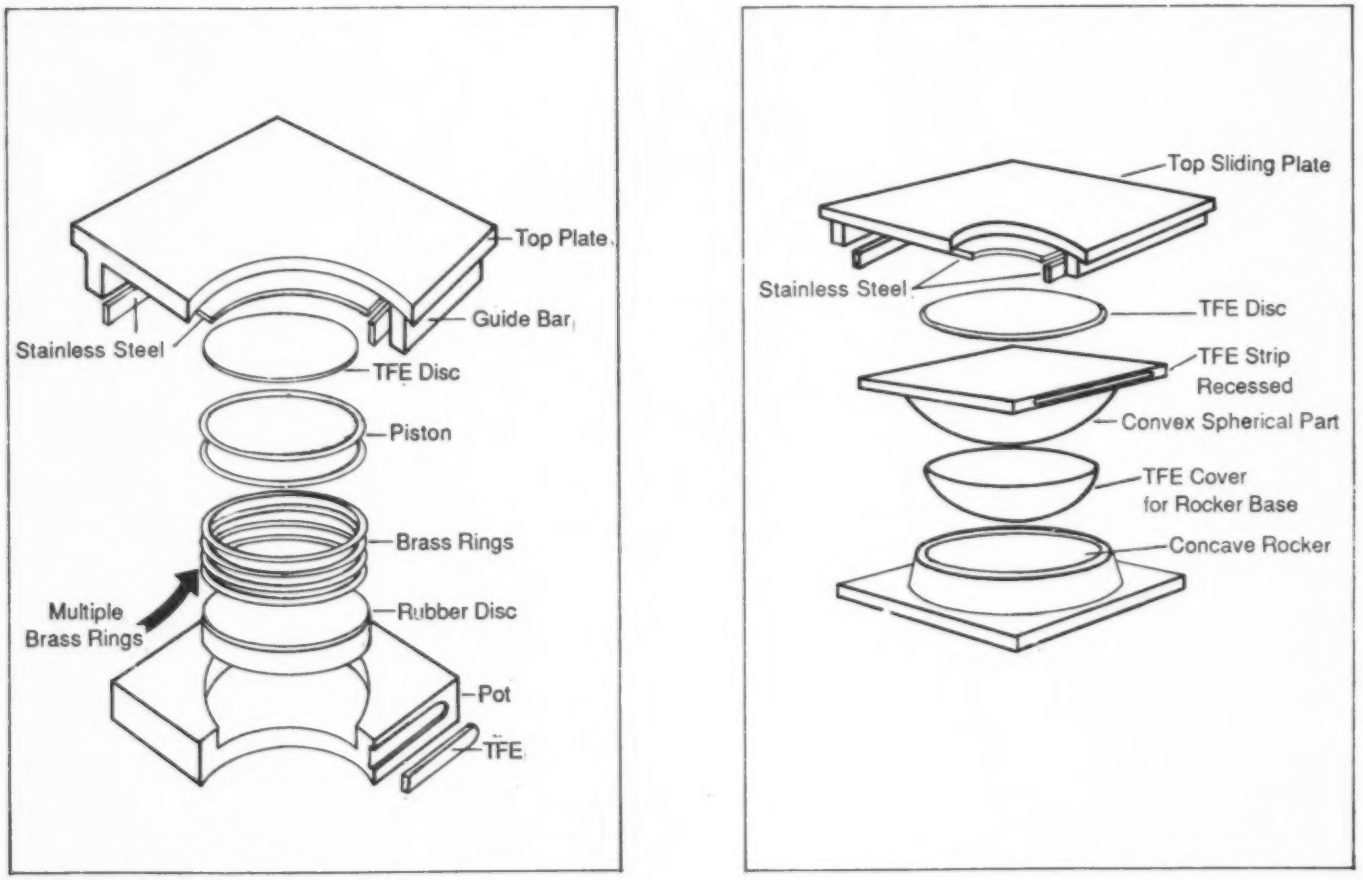
Task 1. A review of domestic and foreign practice and research findings.

Task 2. Analyse and evaluate the information generated in Task 1 and identify those parameters not adequately documented.

Task 3. Recommend a program of laboratory testing.

Task 4. Present the findings of Tasks 1, 2, and 3 in an interim report.

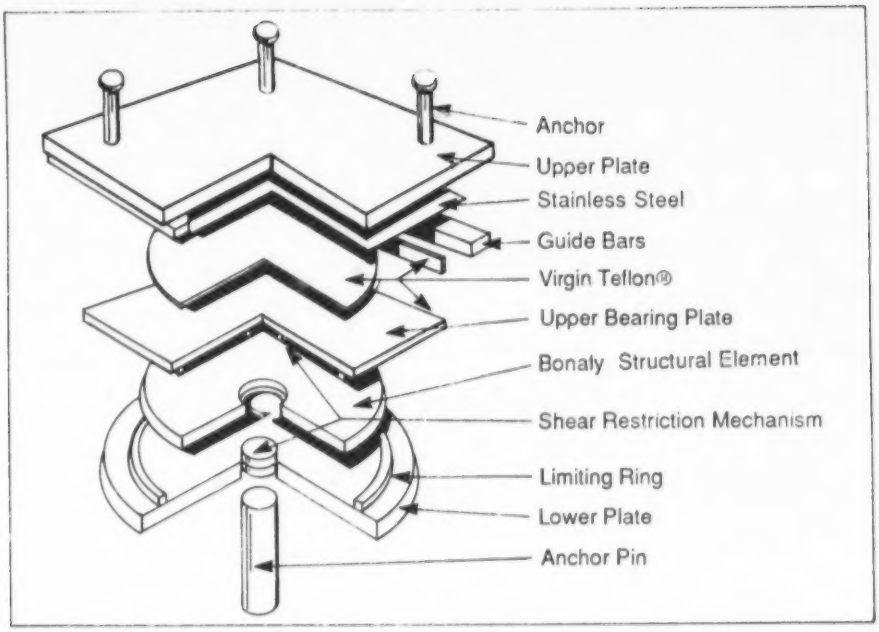
Work on each of the first three Tasks is presented in this report which represents completion of Task 4. Chapter 2 outlines a comprehensive literature survey, by means of which 126 publications relevant to TFE sliding bridge bearings were obtained. The study of this literature reported in Chapter 3 reveals that the coefficient of friction of TFE is of paramount importance in assessing the performance of TFE sliding bearings. Those parameters influencing the friction properties of the TFE sliding surface are identified. Influences which have not been clearly documented, or for which contradictory trends have been reported, are discussed.



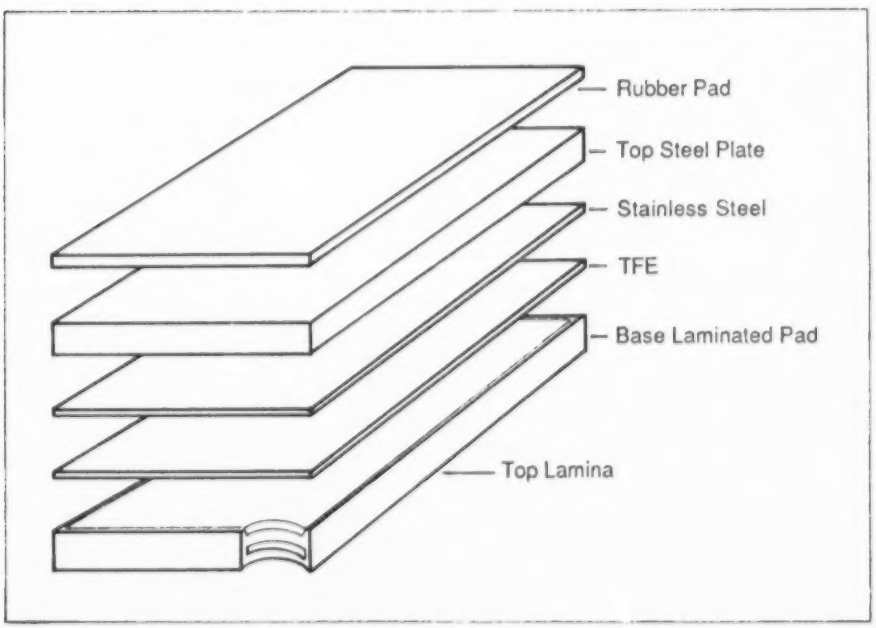
(a) POT

(b) SPHERICAL

FIG.1 POT AND SPHERICAL BEARINGS [1]



(a) DISC [2]



(b) SLIDER [1]

FIG.2 DISC AND SLIDER BEARINGS

2. REVIEW OF DOMESTIC AND FOREIGN PRACTICE

2.1 GENERAL

A comprehensive search of the literature on the use of TFE to provide sliding surfaces in bridge bearings was undertaken as required for Task I. The literature collected is listed alphabetically in Appendix A which contains 126 references. This bibliography has been assembled from two main sources, namely a search of the Transportation Research Information Service (TRIS) database and visits to a number of locations in Europe. Additional information has been obtained by personal contact with the MTC and with bearing suppliers in Canada, namely Goodco Limited [1], Elastometal Limited [2], Wercholoz Canada Inc. [5] and Canadian Industrial Agencies [6]. Catalogues of the bearings supplied by these four companies have been obtained. Further, a number of publications were obtained and useful personal contacts were made during the Second World Congress on Joint Sealing and Bearing Systems for Concrete Structures held in San Antonio, USA on September 28 - October 2, 1986.

2.2 TRIS DATABASE

The searches of the TRIS Database, which were carried out by the MTC, using key words such as TFE, PTFE, tetrofluorethylene, teflon and bridges yielded 60 references. Some references were duplicated in the searches. An examinaton of the titles of these references indicated that 25 appeared to be relevant to this study. Copies of these 25 references were obtained by various means from the sources indicated on the TRIS output. These 25 references are included in Appendix A.

2.3 EUROPEAN VISITS

Visits were undertaken to six locatons in Europe: two were located in West Germany, one in Switzerland and three in the British Isles. A list of places visited, together with names of the personnel consulted, is given in Appendix B. The 33 publications obtained from these visits are included in Appendix A. Many of the publications obtained in West Germany and Switzerland are in German. However, translations to some of these were obtained from various sources. Partial translation of some other documents in German were undertaken when necessary.

2.4 SUMMARY

Appendix A gives a comprehensive list of publications relevant to TFE sliding bridge bearings. Of these 126 publication only 31 relate directly to TFE bearings as used in North America. The visits to Europe confirmed that research and development on TFE bearings are confined primarily to West Germany. Practices in other countries are based largely on German

experience. Further, European experience indicates that cases of unsatisfactory performance of TFE sliding bearings in service are due largely to improper installation.

3. ANALYSIS AND EVALUATION OF INFORMATION

3.1 GENERAL

Literature collected in Task 1 was reviewed and analysed during Task 2. Findings from this analysis are presented in this Chapter, with reference to the parameters which may influence the performance of a sliding surface comprising TFE bearing on a metallic plate. Such parameters are:

(1). coefficient of sliding friction (initial and after wear-in);

(2). effect of dimpling and lubrication;

(3). contact pressure;

(4). speed of travel;

(5). eccentric loading;

(6). temperature;

(7). creep;

(8). roughness of stainless steel or aluminum surface;

(9). different types of TFE (virgin TFE, woven TFE fabric and filled TFE);

(10). method of attachment of the TFE to the backing plate;

(11). surface contamination;

(12). length of the travel path;

(13). load and travel history;

(14). specimen size;

(15). wear.

The first ten parameters are those identified by MTC in the Research Need Statement for this project. The remaining five parameters were identified during Task 2.

3.2 COEFFICIENT OF SLIDING FRICTION

Two coefficients of sliding friction, namely, static and dynamic, exist for TFE sliding on a metallic surface. The presence of both the static coefficient (μ_S), which relates to the force required to start movement, and the dynamic coefficient (μ_D), which relates to the force to maintain movement, are indicated in Fig.3. The static coefficient is usually higher than the dynamic coefficient [8,9,10,11], although conflicting results, indicating the dynamic coefficient as being the higher of the two, exist [12,13,14]. It is also reported [11] that the average difference between the magnitudes of these two coefficients is approximately 0.02 and that the variation in this difference seems to be independent of sliding speed or normal pressure at the TFE/metallic surface interface. Figure 3 indicates that the difference between the two coefficients appears to reduce with increasing cycles of movement.

The initial friction level is critical as the initial movement due to, for instance initial post-tensioning or stripping of formwork, occurs generally before the concrete has attained its specified strength. The initial peak value of the frictional coefficient can be reduced significantly by running-in the bearing, using lubrication on the sliding surfaces [15,16] or using a highly polished metallic sliding plate [8]. Running-in a bearing would necessitate subjecting it to a number of cycles of movement prior to installation. The reduction in the coefficient of friction with increasing number of cycles of movement in an unlubricated surface is obvious in Fig.4. Figure 4 also shows that the initial coefficient of friction is reduced by a factor of 4 by the addition of lubrication and by a factor of 3 when the surface finish on the stainless steel plate in a TFE and stainless steel interface is improved from a standard finish to a highly polished mirror finish. Surface finish of the TFE does not appear to have a significant effect on the friction, presumably due to the surface being soft enough to adapt itself by deforming [17].

As can be seen in Fig.5, due to the higher value of static coefficient of friction, the phenomenon of "stick-slip" occurs, particularly during the first movement. This phenomenon is characterised by short periods of slowly increasing frictional force followed by a rapid fall in the force.

The two coefficients of sliding friction are the prime parameters in the design of a TFE sliding bridge bearing, since they dictate the horizontal force transmitted from the superstructure to the substructure. Consquently, factors affecting the coefficient of friction are of paramount importance.

The coefficient of friction may be influenced by the parameters identified as (2) to (15) in Section 3.1 of this report. Consequently, further discussion is concerned primarily with the influences of these parameters on the coefficient of sliding friction in a TFE/metallic surface interface.

3.3 DIMPLING AND LUBRICATION

Dimpling of the TFE sliding surface, to facilitate lubrication, is used in Canada and in Europe. A typical layout and the dimensions of the dimples are shown in Fig.6. The area of the

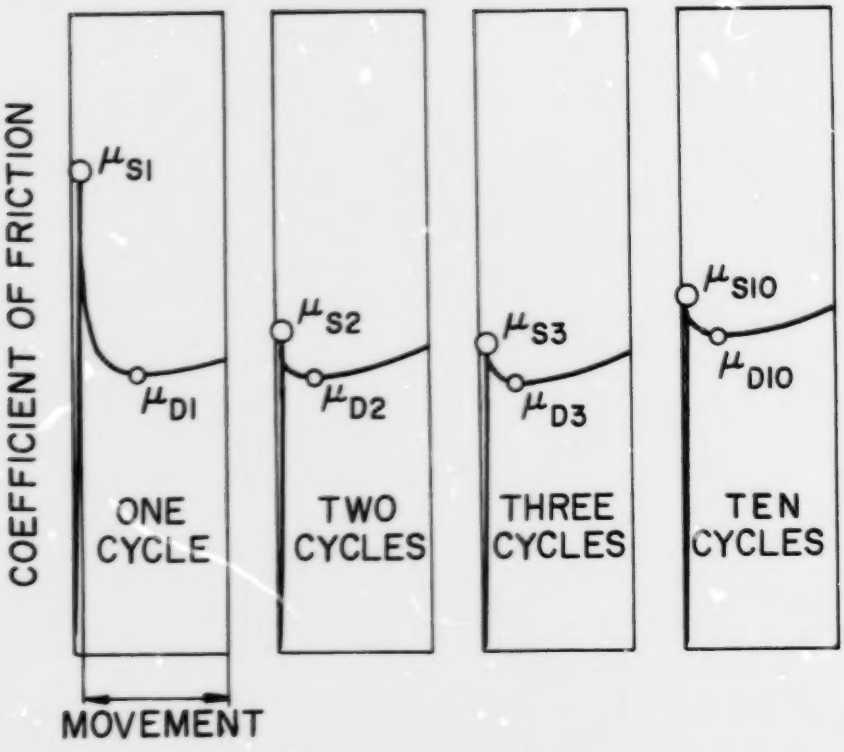


FIG.3 VARIATION OF THE COEFFICIENT OF FRICTION WITH LENGTH OF TRAVEL [7]

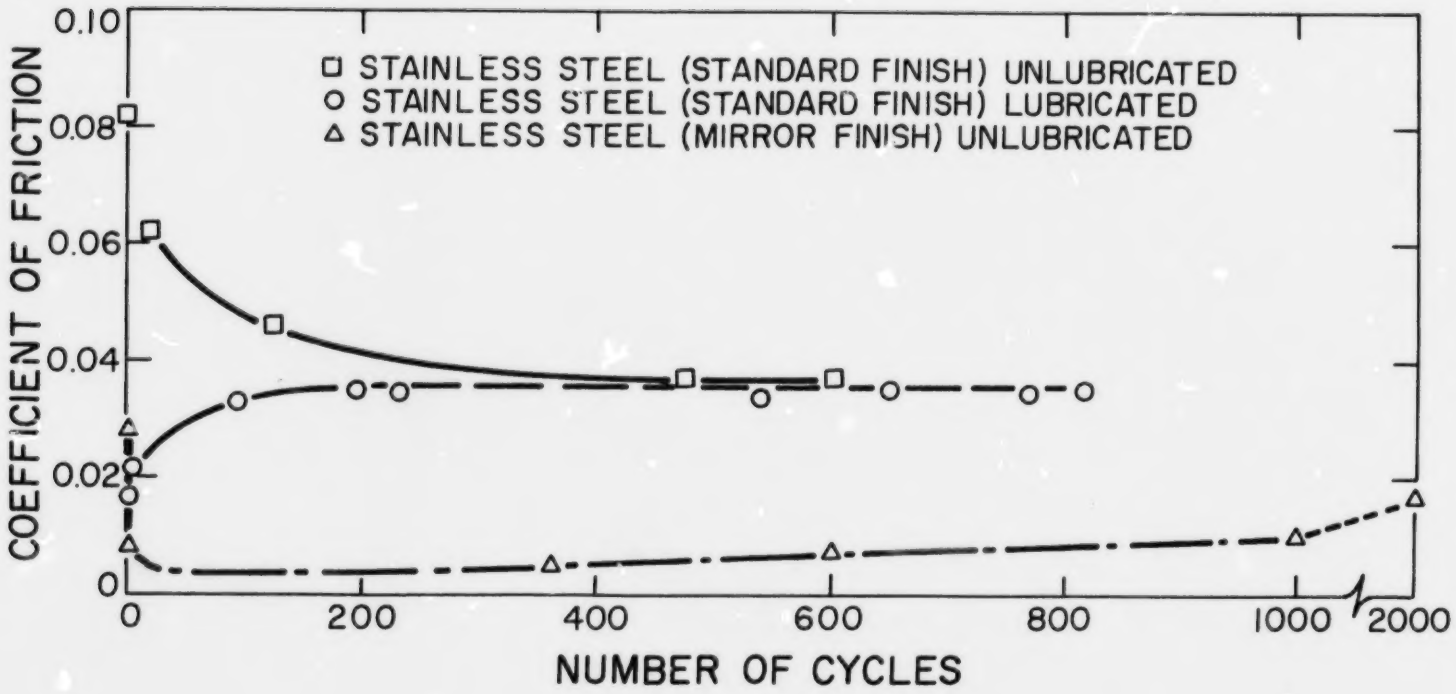


FIG.4 COEFFICIENT OF FRICTION VS. NO. OF CYCLES [8]

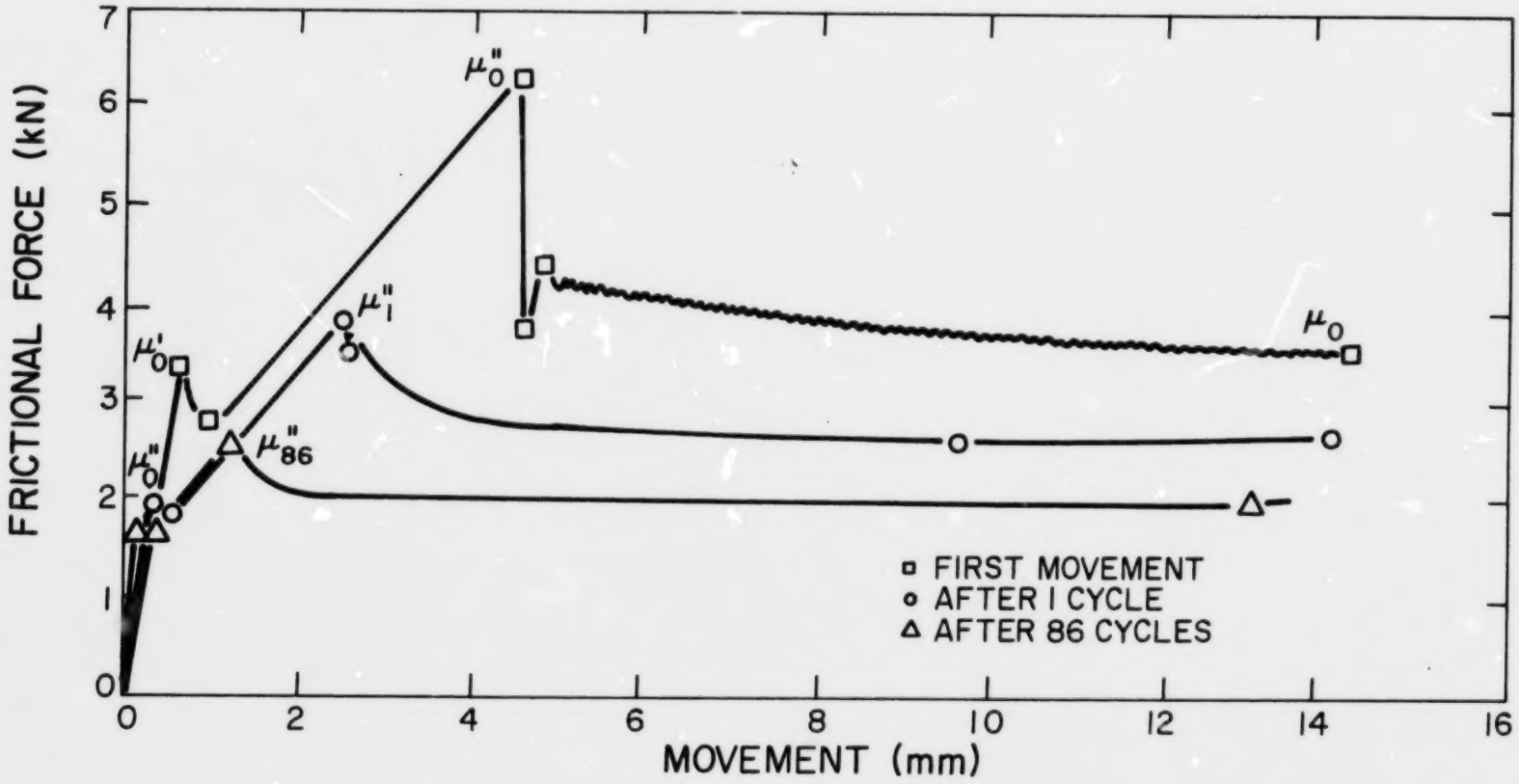


FIG.5 FRICTIONAL FORCE VARIATION WITH MOVEMENT OF SLIDING PLATE [9]

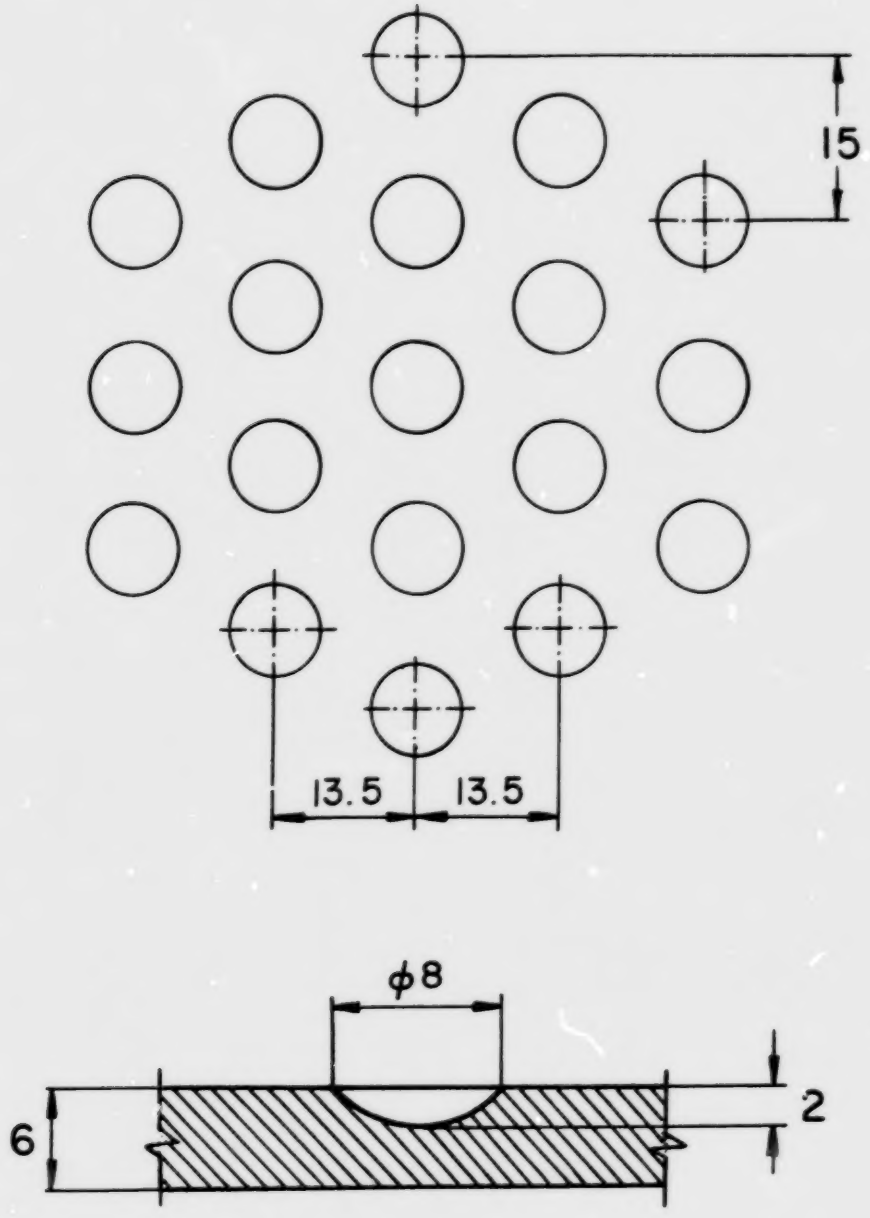


FIG.6 DIMENSIONS AND ARRANGEMENT OF GREASE DIMPLES [18]

dimples should be between 20% and 30% [19] or 10% and 30% [20,21] of the total surface area. Figure 7 shows the beneficial effects of both lubrication (compare curves I and II) and the use of dimples (compare curves II and III) on the coefficient of friction over a slide path of 1 km. It can be seen that lubrication reduces the initial coefficient of friction significantly and that provision of dimples extends the effective life of the lubricant. The dimples serve as a reservoir for storage of the lubricant which is under hydrostatic pressure and is extruded to the sliding surface under pressure. Silicone greases are recommended as suitable lubricants [22]. They prove to be effective at a temperature as low as -35 °C, and will not resinify nor attack the material of the sliding plates. The Ontario Specification for bearings [19] requires that a silicone grease lubricant conforming to a US Military Specification [23] be used.

Replacement of the lubricant on a conventional dimpled TFE surface (Fig.6) of a bearing in service is difficult. An alternative layout [24] for the TFE surface, aimed at permitting re-lubrication in service, has been developed in Switzerland and is shown in Fig.8. The sliding surface comprises individual TFE discs, each containing an inter-connecting pattern of surface grooving and recessed in a steel backing plate to form intermeshing annular sliding chains. The large lubricant storage spaces between the individual sliding chains ensure prolonged functioning of the sliding bearing. Relubrication of the sliding surface can be made via a hole leading from the edge of the backing plate into the lubricant storage spaces. However, conflicting opinions [18] have been expressed as to the feasibility of relubrication of a bearing in service, since the grease tends to harden in the long term, making ejection of the old grease by application of the new extremely difficult. Hardening of the lubricant on a dimpled surface has been observed in bearings taken out of service after fifteen years by researchers at the RWTH Aachen in West Germany.

It is noteworthy that, in compliance with American Standard Specifications [25,26], bearing manufacturers in the United States do not use lubrication or a dimpled surface in TFE sliding bearings. This may be due to the fact that the effect of lubrication without dimples is limited in the long term (see Fig.7) and also due to the difficulty of relubricating a bearing in service.

The beneficial effects of both lubrication and dimpling are well documented. Lubrication reduces the coefficient of friction while dimpling prolongs the effective life of the lubricant.

3.4 CONTACT PRESSURE

It is recognized that the coefficient of friction decreases with increasing contact pressure in the TFE/metallic plate interface. Figure 9 shows the variation of the coefficient of friction with pressure at a temperature of +20 °C for both lubricated and unlubricated specimens. The trend is similar in each of the two cases. The coefficient of friction reduces by approximately 40% for the indicated increase in contact pressure for both cases. Figure 10 shows the variation of the coefficient of friction and related frictional force with increase in pressure for an unlubricated specimen. It should be noted that, although the coefficient of friction decreases with increasing pressure, the frictional force increases. Therefore, in service the maximum frictional force in the bearing will always be associated with the maximum loading.

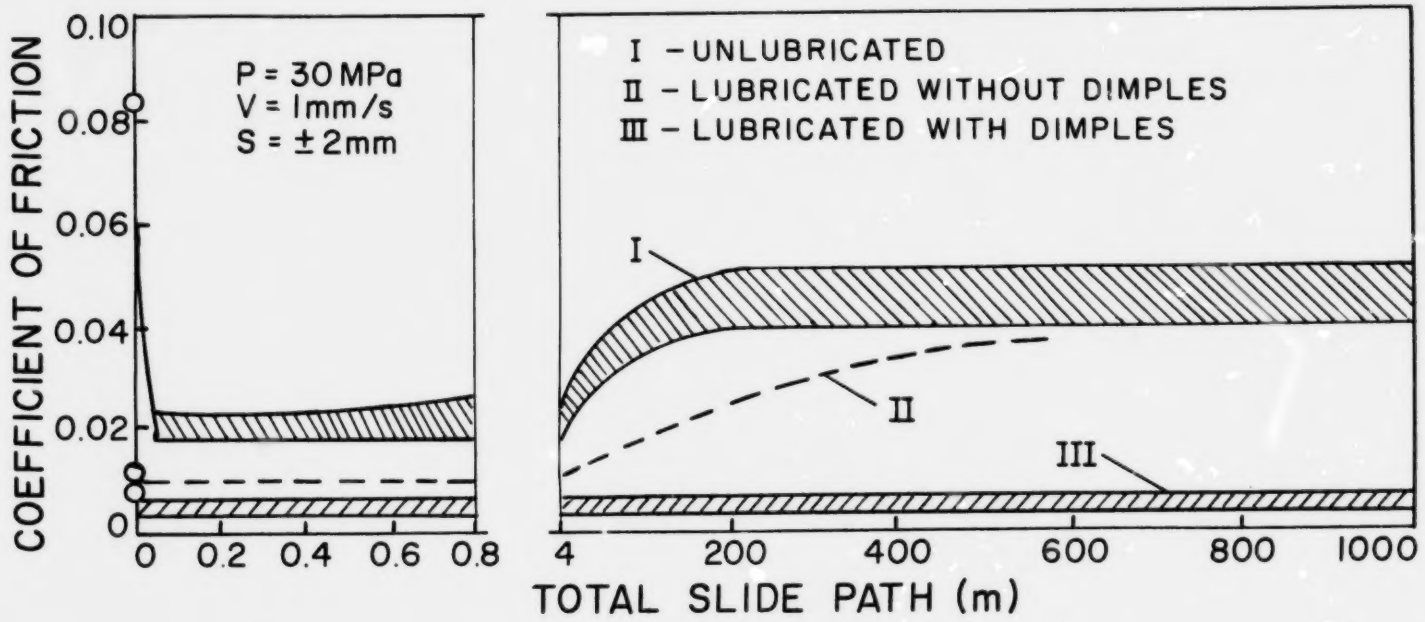


FIG.7 INFLUENCE OF LUBRICATION AND DIMPLING ON THE COEFFICIENT OF FRICTION [18]

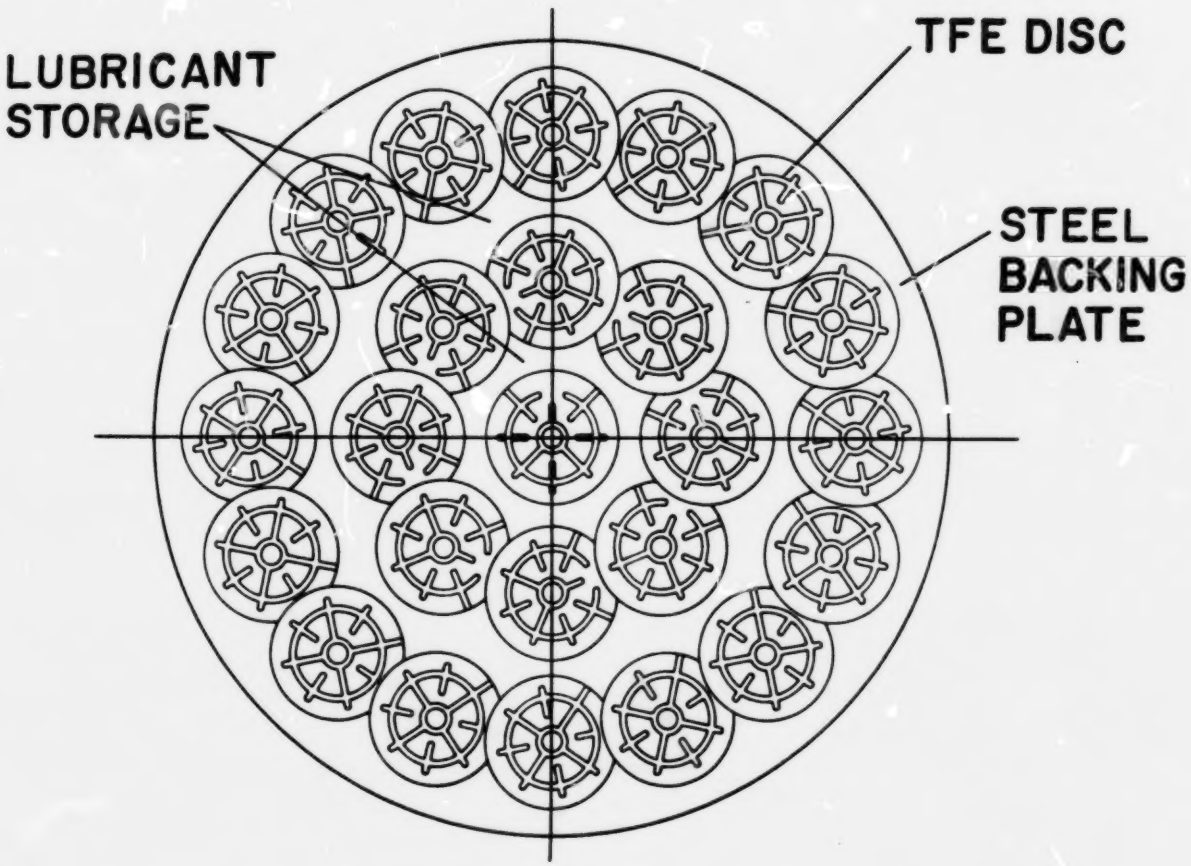


FIG.8 TFE DISC SLIDING CHAINS [24]

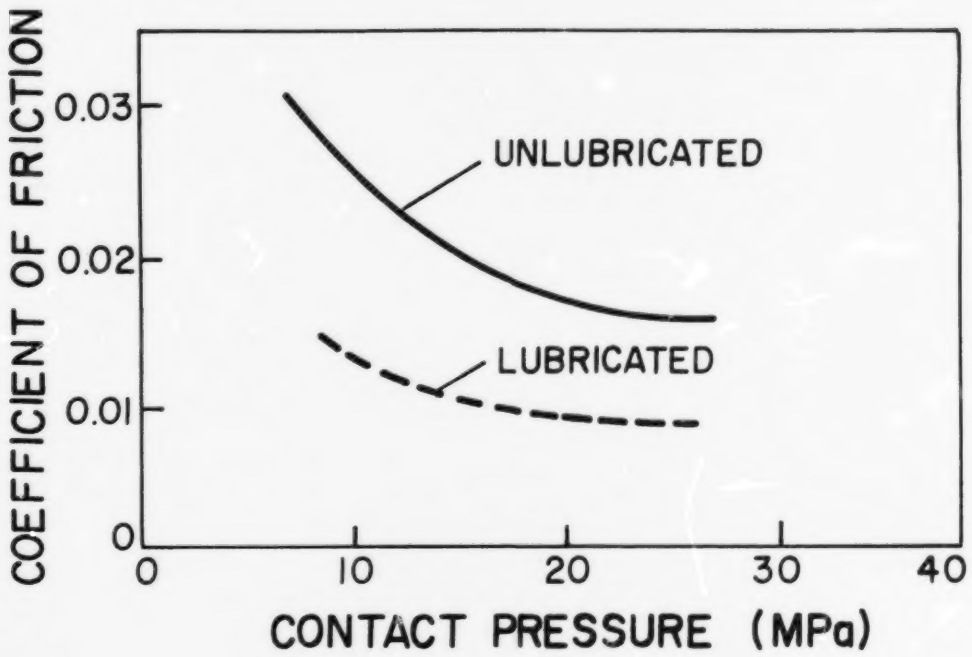


FIG.9 INFLUENCE OF CONTACT PRESSURE AND LUBRICATION ON THE COEFFICIENT OF FRICTION [15]

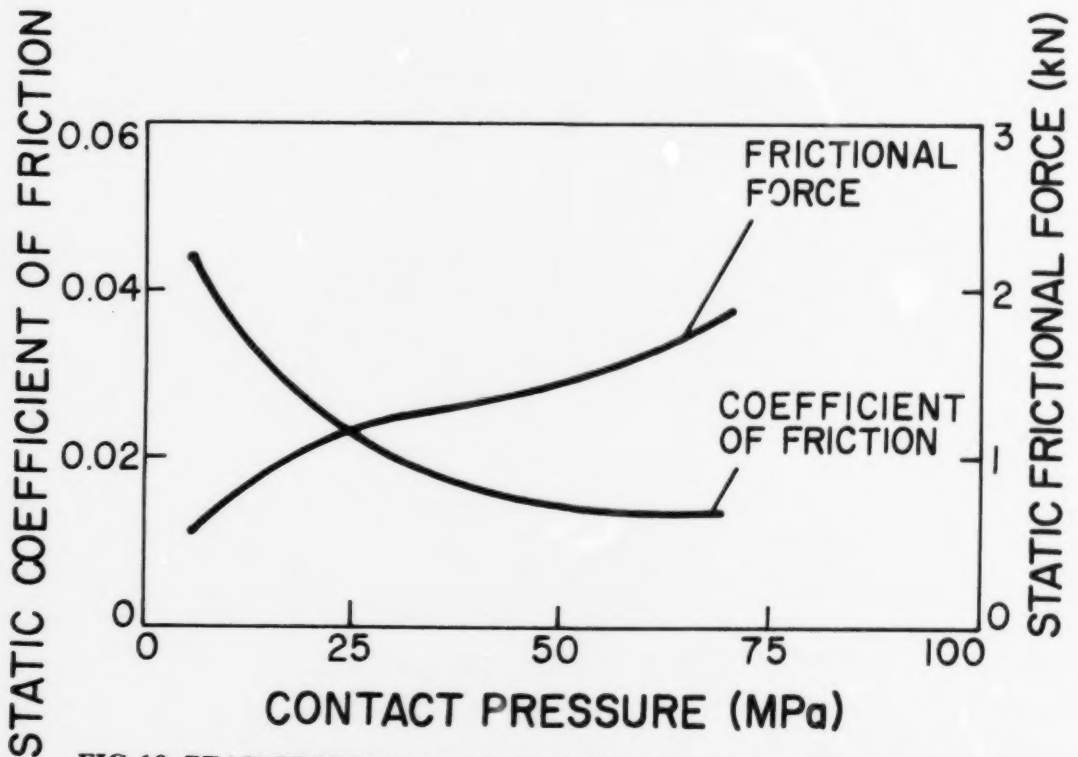


FIG.10 PEAK FRICTIONAL FORCE AND COEFFICIENT OF FRICTION AS A FUNCTION OF CONTACT PRESSURE [8]

Some studies [12,27] have indicated that, after the contact pressure reaches about 14 MPa, any further increase in pressure does not reduce the coefficient of friction significantly. Figure 11 shows that the variation of frictional coefficient has a tendency to level off when contact pressure is close to 10 MPa, whereas Fig.10 indicates that this levelling off occurs only at presures in the order of 50 MPa. It appears from Fig.9 that levelling off may occur at lower pressures when lubrication is used.

The phenomenon that the value of frictional coefficient levels off at a certain level of contact pressure may be explained using the elastic and plastic deformation theories of plastics [8,29,30]. At low pressures, where deformation has been shown to be elastic, the real area of contact, A, of the surface is given by

$$A = kN^n \tag{3.1}$$

where k is a parameter depending upon the assumed form of the surfaces and the elastic constants of the materials, N is the normal applied load and n has a value between 2/3 and 1 depending upon the form of the surfaces.

At high loads, the deformation changes from elastic to plastic, and the area of contact, A, which is in proportion to the load, N , is given by

$$A = N/p_y \tag{3.2}$$

where p_y is the compressive yield strength of TFE.

The coefficient of friction, μ, is expressed as

$$\mu = F/N \tag{3.3}$$

where F is the frictional force which is given by the product of interface shear strength, τ, and the real contact area, A, of the TFE.

The interface shear strengh is unlikely to be dependent on the load level and thus the real area of contact is the major factor influencing friction. At higher loads with plastic deformation, the frictional force F will increase in direct proportion to N, since

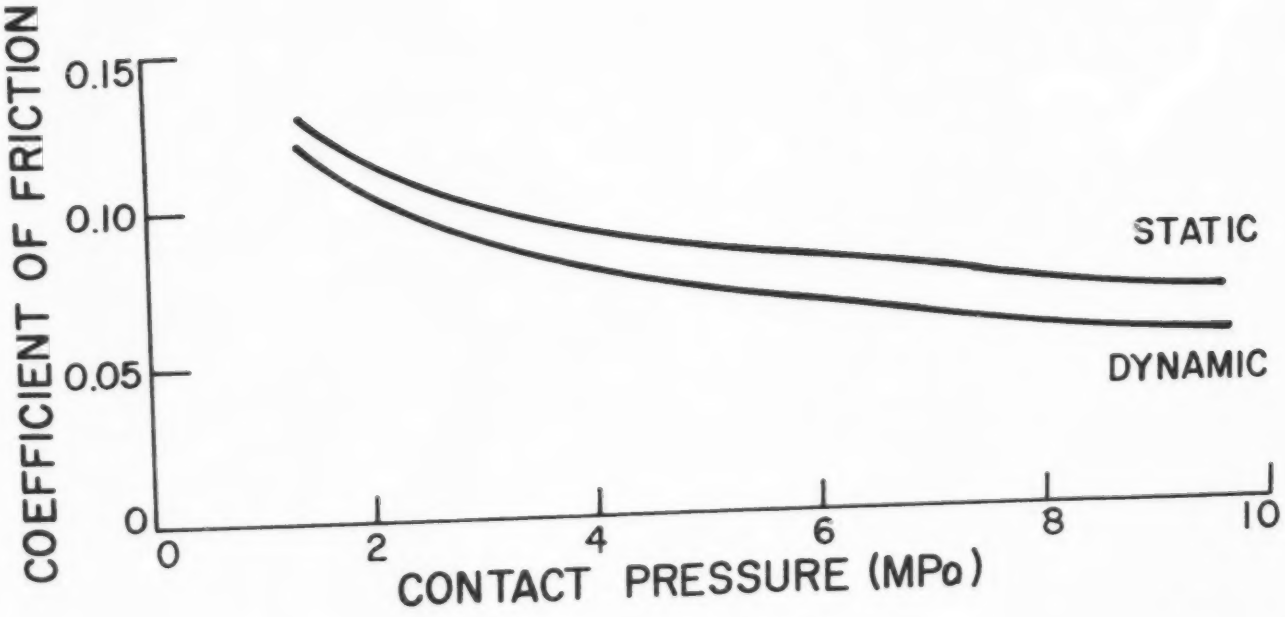


FIG.11 STATIC AND DYNAMIC COEFFICIENTS OF FRICTION VS. CONTACT PRESSURE [28]

$$F = \tau A = \tau N / p_y \tag{3.4}$$

and thus the coefficient of friction will be constant.

Under elastic conditions at lower loads, the friction force is given by

$$F = \tau A = \tau k N^n \tag{3.5}$$

Therefore, the coefficient of friction which is given by

$$\mu = F/N = \tau k N^{n-1} \tag{3.6}$$

will decrease with an increase in the applied load within the elastic range since $n < 1$.

Equation (3.6) can be written as

$$\begin{aligned} \mu &= \tau k A_0^{n-1} (N^{n-1} / A_0^{n-1}) \\ &= \tau k A_0^{n-1} P^{n-1} \\ &= Q P^{n-1} \end{aligned} \tag{3.7}$$

where P is the average pressure at the bearing sliding surface, A_0 is the geometrical area of the contact surface, $Q = \tau k A_0$ is a function of the interface temperature and sliding speed [8,31] and n is a function of filler content of the TFE, surface lubrication and surface finish of the metallic sliding plate. Test results [8] indicate that, for pressure not exceeding 40 MPa, Q is equal to 0.15 and 0.28 for a sliding speed of 0.022 mm/sec at 20 °C and -5 °C, respectively, and n is 0.58 for an unlubricated stainless steel plate with a surface finish of 0.4 μm CLA sliding on unfilled TFE. The terms CLA and fillers for TFE are discussed later in Sections 3.9 and 3.10, respectively.

It has also been reported [9] that, at pressures above 25 MPa, "stick-slip" is no longer detectable during any cycle except the initial one, indicating that the difference between the static and dynamic coefficients of friction diminishes with increasing pressure level.

In order to achieve satisfactory low frictional coefficient values, the contact pressure should exceed 3 MPa [17]. The allowable average contact pressures on the TFE in bearings in

service are normally 30 MPa under dead load and 45 MPa under total load [3,19,20,21,32] with the exception of 24 MPa under total load in the United States [25,26].

It is well documented that the coefficient of friction decreases with increasing pressure and approaches a minimum value at a certain pressure. The magnitude of this pressure has been reported to vary between 10 MPa and 50 MPa. The trend in reduction of the coefficient of friction with increasing pressure appears to reflect the change from elastic to plastic behaviour in the TFE.

3.5 SPEED OF TRAVEL

At low sliding speed in the TFE/metallic interface, the coefficient of friction increases with sliding speed as indicated in Fig.12 and also by laboratory tests [33] conducted at sliding speeds of 0.4mm/sec and 2.0 mm/sec. Figure 13 shows that, the coefficient of friction increases with sliding speed up to a speed of about 1000 mm/sec [29,34] and then levels off. One possible explanation [29] for this behaviour may be due to the combined effects of "stick-slip" and shearing of the TFE.

At low speeds, the static friction portion of stick-slip is much in evidence with a corresponding significant effect in producing large friction values. Also in the low speed range, when speed is increased, the amount of shearing within the TFE increases. Thus the increase in friction in the low speed range can be a result of increased shearing, with stick-slip contributing its share to the total friction value. In the high speed range the static friction component approaches the average dynamic one and also shearing occurs nearer the surface of the TFE. This shearing has a reduced effect on the total friction value resulting in a levelling off in friction at high speeds.

These trends have been investigated by a microscopy study [35] of unlubricated TFE. This study showed that, at low sliding speed and moderate temperature, a thin film of TFE is drawn over the sliding surfaces. This film has a highly oriented crystal structure and easy shear, aided by creep, of the TFE crystals results in a low friction value under this condition. Low friction values obtained from field measurements on bearings in Switzerland [31,36] are attributed to the influence of TFE creep in the horizontal direction at the low sliding speeds experienced in service. At high speed, transfer of relatively large fragments of TFE in the form of lumps, ribbons or sheets occurs. This is a result of the viscous force to shear this film increasing until a stage is reached where the shear force exceeds the strengh of the boundaries between the non-oriented crystals of TFE. This results in transfer of lumps at least as thick as a single crystal of TFE and gives rise to a higher coefficient of friction.

Little information is available on the sliding speeds experienced by bearings in service. Thermal movements produce negligible sliding speeds compared to those induced by traffic on a bridge. Rates of movements up to 0.012 mm/h/m and 0.07 mm/h/m (or 0.0003 mm/sec and 0.002 mm/sec per 100 m) have been recorded due to change in temperature for a reinforced concrete bridge deck and an unsurfaced steel box structure, respectively [8]. Based on these extremely slow speeds, it has been suggested that only static friction should be considered [15]. However, field measurements show maximum sliding speeds at bearings due to traffic load as

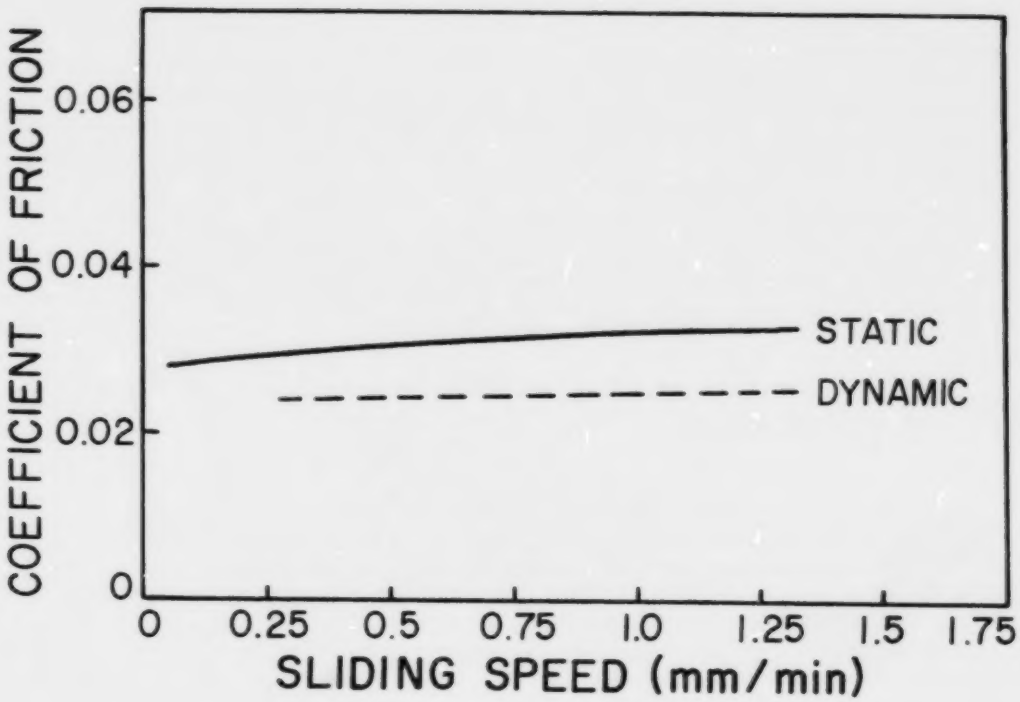


FIG.12 STATIC AND DYNAMIC COEFFICIENTS OF FRICTION VS. SLIDING SPEED [8]

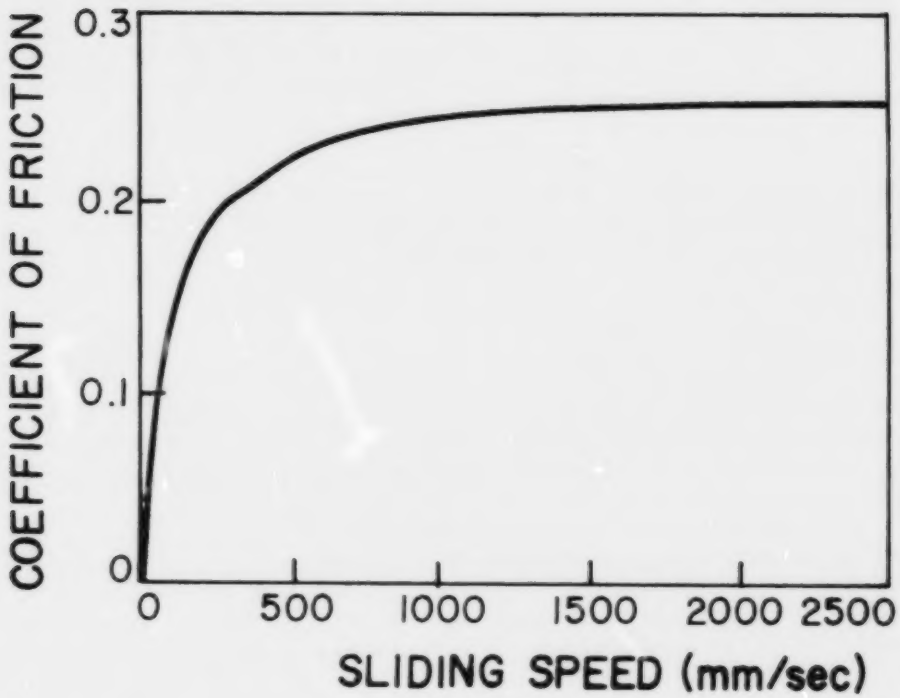


FIG.13 COEFFICIENT OF FRICTION VS. SLIDING SPEED [34]

high as 20 mm/sec for a steel railway bridge [18] and 70 mm/sec for a steel/concrete composite bridge [37]. These values are well below the speed of 1000 mm/sec at which the coefficient of friction becomes independent of speed and thus laboratory tests should be undertaken at speeds of up to 70 mm/sec. The maximum speed used in laboratory tests to date appears to be 2.5 mm/sec [15].

The speed of travel is an important parameter which must be addressed when specifying the coefficient of friction in a TFE sliding bearing. Field measurements are required in order to establish actual sliding speeds for particular bridge types.

3.6 ECCENTRIC LOADING

Bridge superstructures are flexible and the deflections which occur under load must be accommodated by rotation at the bearings. The most common types of modern bearings are shown in Figs.1 and 2. Accommodation of the rotation of the superstructure relative to the substructure results in eccentric loading patterns within the components of the bearing. Eccentricity produces moment which induces additional edge pressure on the TFE. Reduction of pressure on an edge may cause uplift resulting in the TFE being squeezed out, while increased pressure may cause flattening of the TFE and closure of the grease dimples. This will impair the performance and reduce the life of the bearing through increased wear of the TFE.

Increased wear was observed at the edges of the TFE in a pot bearing taken from service after 15 years by researchers at RWTH Aachen in West Germany. X-ray pictures of the TFE indicated the presence of metallic particles in the grease in the dimples near the edge of the TFE but not elsewhere. Further, tests on this bearing, using pressure sensitive paper inserted between the TFE and the steel sliding plate, indicated that the eccentricity present in a pot bearing may be greater than the traditionally assumed value of 4 percent of the internal diameter of the pot. In a spherical bearing the eccentricity returns to zero once rotation has taken place.

The authors are not aware of any test data relating specifically to the effect of eccentricity of load on the coefficient of friction of TFE and consequently some tests should be carried out in this area.

3.7 TEMPERATURE

The value of the coefficient of friction is affected by temperature. Figure 14 shows the variation of the coefficient of friction with temperature after a total sliding path of 10 km in a test [18] where the pressure on the TFE was 45 MPa, the sliding speed was 0.4 mm/sec. and the roughness of the metallic surface, R_{max}, was 1 μm (R_{max} is defined later in Section 3.9). The coefficient of friction increases with decreasing temperature, the rate of increase becoming larger at temperatures below -20 °C. It can be seen that the coefficient of friction increases by a factor of 4 as the temperature drops from +21 °C to -35 °C. Contrary to above, it has been claimed [12], that the coefficient of friction of TFE is virtually independent of temperature over

the range -29 °C to 260 °C, while another investigation [38] reports that the coefficient of friction increases with temperature over the range 20 °C to 150 °C.

Figure 15 shows that variation of the frictional coefficient with temperature change has different trends for surfaces with and without lubrication. The rate of increase in the coefficient of friction with decrease in temperature is significantly lower for the lubricated surface. A microscopy study [35] indicates that the phenomenon of transfer of large fragments of TFE and resulting high coefficient of friction observed at high sliding speeds occurs also at low temperature.

Conditions at the TFE/metallic surface interface in a bearing may be modified quite significantly by condensation and icing which may occur at low temperature [8]. Condensation may act as a lubricant at the interface and reduce the coefficient of friction. Ice formation on the sliding plate will create additional restraining forces, which have to be overcome before sliding can occur, resulting in an apparent increase in friction.

The contradictory opinions regarding the variation of frictional coefficient with temperature change need to be clarified. The behaviour of lubricated TFE sliding bearings at low temperature is of interest for applications in Ontario and it is recommended that testing in this area be undertaken

3.8 CREEP

TFE is a visco-elastic material and exhibits creep under sustained pressure in excess of 7 MPa. By confining the TFE sheet within a sharp edged recess [18] and limiting the maximum contact pressure level, a significant reduction in both the magnitude of the deformation and the time period to reach steady state conditions can be obtained. The reduction in creep due to confinement is demonstrated in Fig.16. Confined TFE is used in bridge bearings. It is suggested [8] that limiting the maximum applied compressive pressure to 35 MPa effectively eliminates the possibility of inducing long term deformation capable of seriously affecting bearing performance. In this case, the maximum deformation will be of the order of one-sixth of the unconfined thickness of the TFE layer. The thickness of the TFE in a bearing should be between 4.5 mm and 6 mm, depending on its overall dimension, with 2.5 mm to 3 mm being recessed [9,19,20,21,32]. Filled TFE and woven TFE fabric (see Section 3.10) exhibit an improved resistance to creep [6,8,11,28].

It appears that creep of the TFE in a bearing can be controlled effectively by limiting the contact pressure and recessing the TFE sheet into its backing plate.

3.9 ROUGHNESS OF STAINLESS STEEL (OR ALUMINUM) SURFACE

A number of different parameters are used to define surface roughness. The CLA (centre line average) measure of surface roughness is synonymous with the arithmetical average. Both

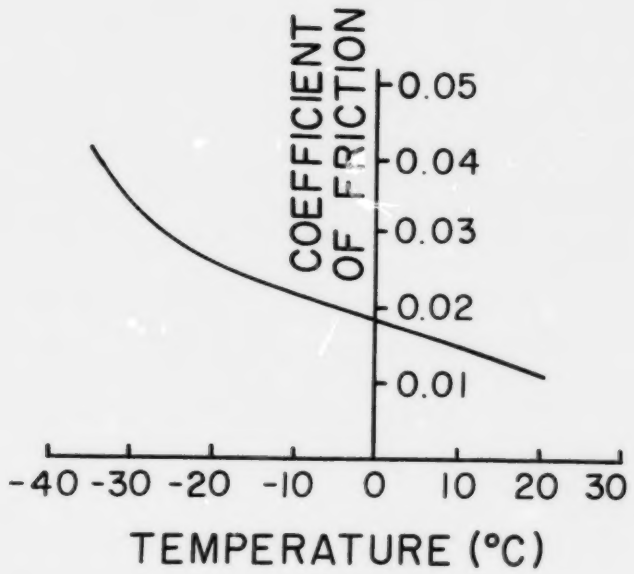


FIG.14 COEFFICIENT OF FRICTION VS. TEMPERATURE [18]

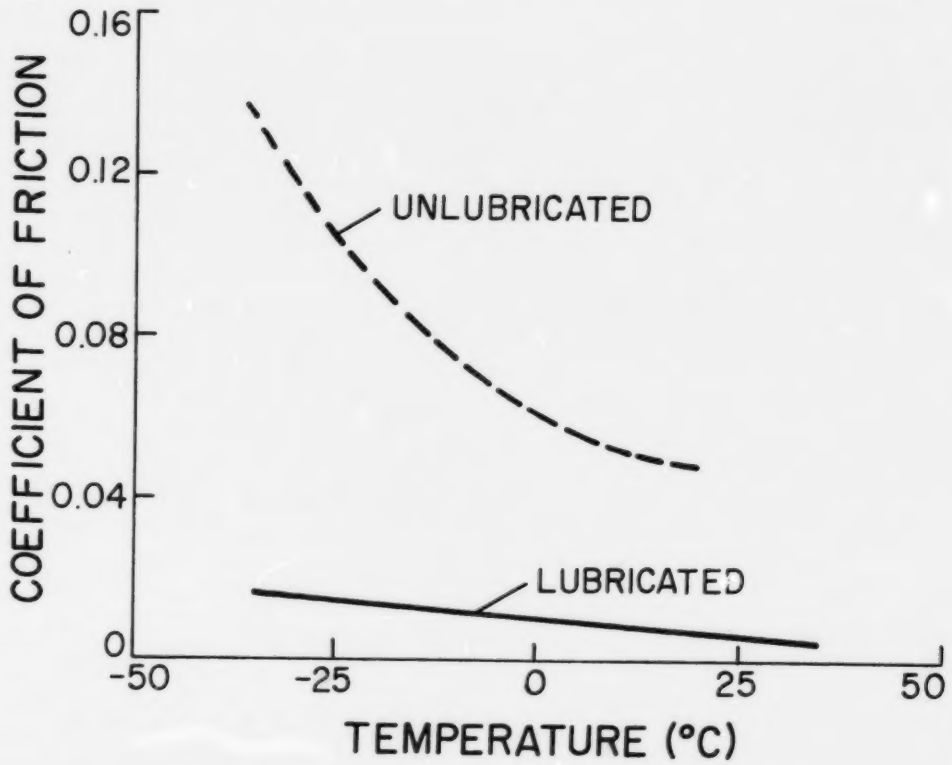


FIG.15 EFFECT OF TEMPERATURE AND LUBRICATION ON THE COEFFICIENT OF FRICTION [39]

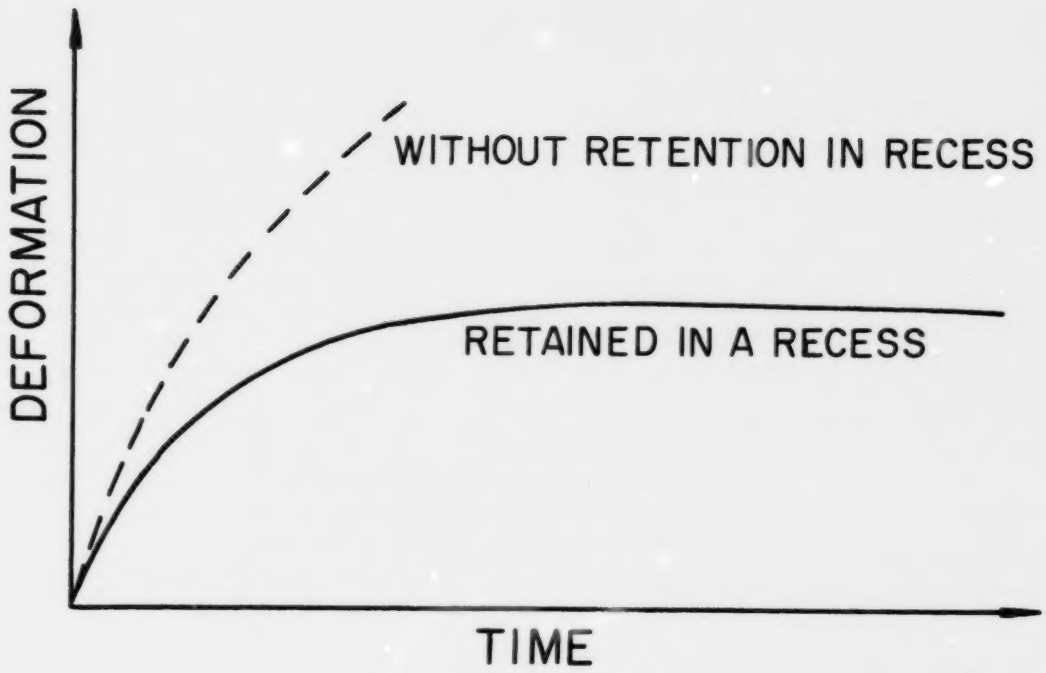


FIG.16 CREEP OF PURE TFE [18]

arithmetical average and root mean square average, which is approximatly 11% higher than the arithmetical average, are used in North America [40,41] to define surface roughness. German codes [42,43] use three parameters R_a, R_z, and R_{max}, while a British code [44,45] uses only R_a and R_z to define surface roughness. The parameter R_a, previously known as CLA, is a measure of the average height of the irregularities on the surface. R_z, previously known as PVA (peak valley average), is a measure of the average total height of the surface irregularities. R_{max} is defined as the maximum value of R_z. Values of R_a and R_z are related according to the chart given in Fig.17.

The sliding surface in contact with the TFE is usually stainless steel although anodized aluminum may be used for spherical surfaces [19,24]. The roughness of the metallic surface influences the coefficient of sliding friction at the TFE interface without lubrication as shown in Fig.18. Initially the coefficient of friction for a highly polished surface (2 μin CLA) is lower than that for a less polished surface (30 μin CLA), but, after a travel path of about 10 m, increases to approach that for the less polished surface. This indicates that after the initial running-in period, the surface roughness of the mating plate is not very significant [10]. This trend is probably due to the phenomenon of TFE transfer to the mating surface.

After a certain travel path in an unlubricated interface, transfer of TFE takes place in the full area covered by the TFE bearing surface, and eventually reaches an equilibrium state where the loss of TFE due to abrasion balances the TFE being transferred. The TFE is then virtually sliding on its thin-layered deposits and slightly different surface finishes do not appear to influence the value of the frictional coefficient significantly. On the other hand, a dimpled sliding surface with lubrication shows practically no abrasion. Tests [22,33] reveal only a few traces of wear or local TFE transfer after a 5 km slide path. Therefore, whether the trend in variation of the frictional coefficient indicated in Fig.18 for different surface roughness without lubrication will occur when a lubricant is used remains unclear.

Curved TFE elements are used in spherical surfaces as shown in Fig.1. Since there are difficulties in machining a curved metallic surface, the finish is generally inferior to that achieved on a flat surface, and as a result non-uniform load distribution, or even load concentration, can occur in the interface in a spherical bearing. Higher pressure reduces the coefficient of friction and generally speaking, under the same average pressure, the coefficient of friction in a planar interface is higher than in a curved one.

The roughness of the metal surface should be less than 0.25 μm and 0.5 μm arithmetic average for planar and curved surfaces, respectively, according to Ontario Specifications [3,19], whereas 0.15 μm and 0.5 μm are stipulated in some other Specifications [20,21]. In the United States a maximum value of 0.5 μm root mean square average is used for the stainless steel mating surface [25,26]. It has been pointed out [10] that different surface roughness values may be obained from the same sample depending upon the direction of measurement, a higher value resulting from measurement across the direction of planing as opposed to parallel to it. Therefore, where the final finishing is emphasized in one direction, this direction should be indicated, so that the mating sheet may be correctly orientated in the bearing.

Surface roughness values produced by common production processes are given in Table 1. Finishes available on sheet products, as defined in ASTM A480 [46], are given in Table 2.

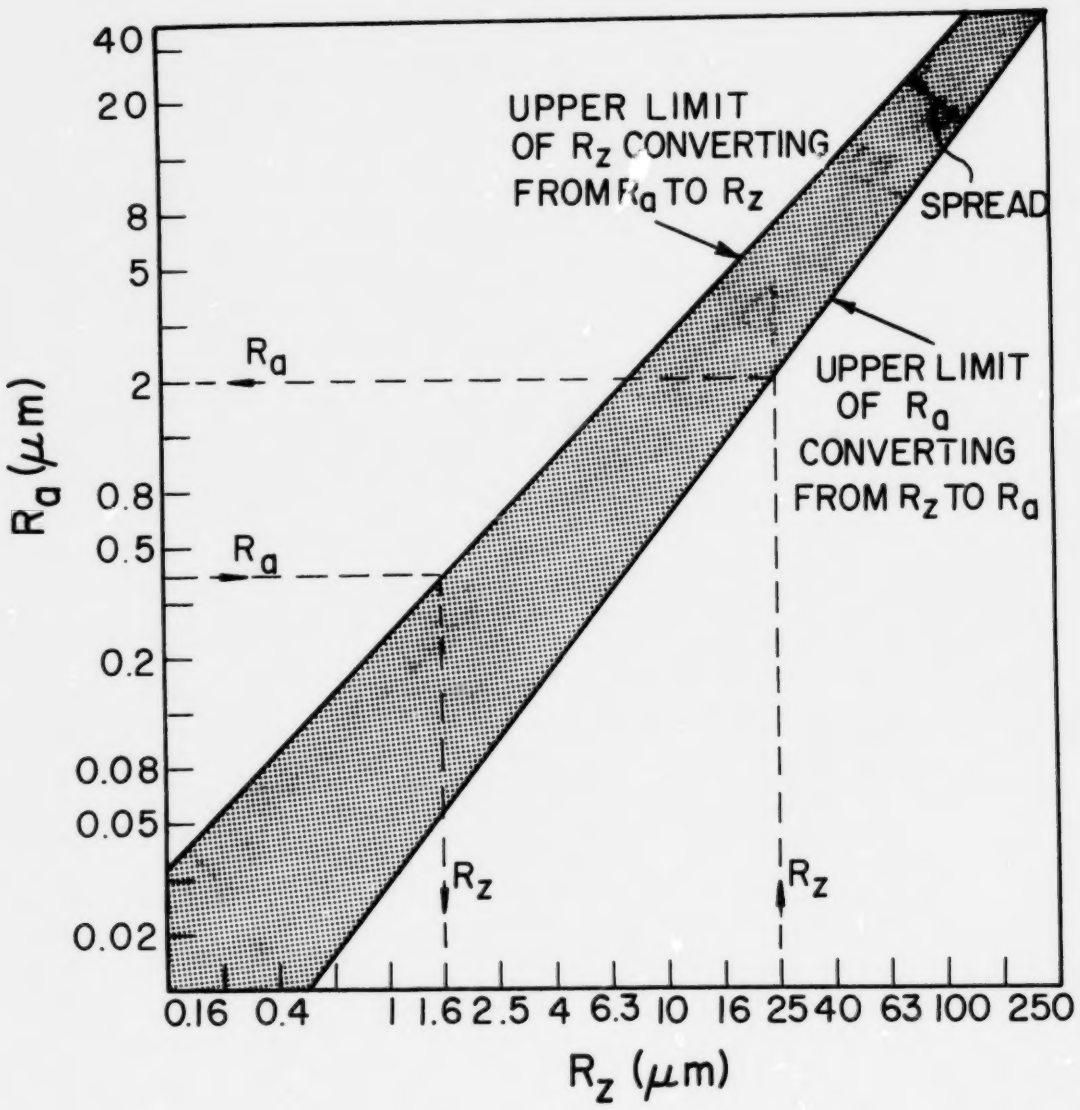


FIG.17 RELATIONSHIP BETWEEN R_a AND R_z [43]

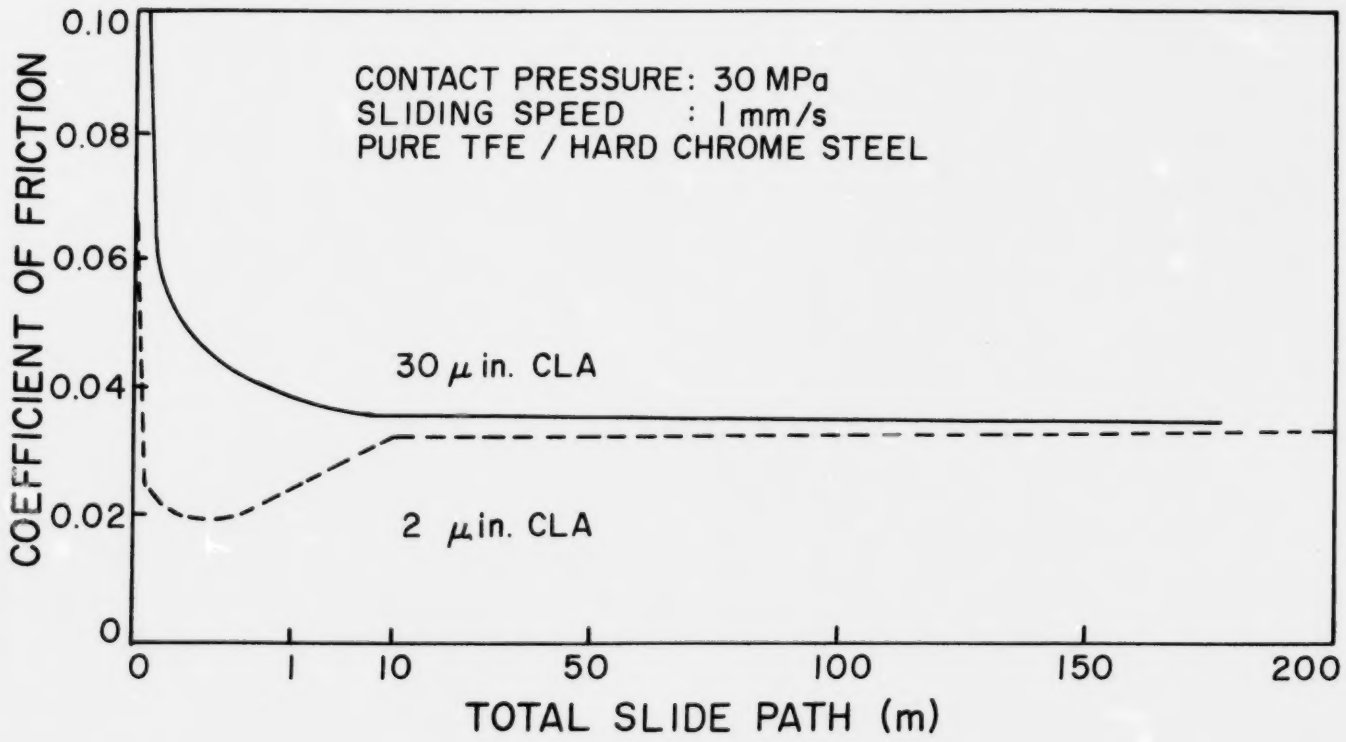
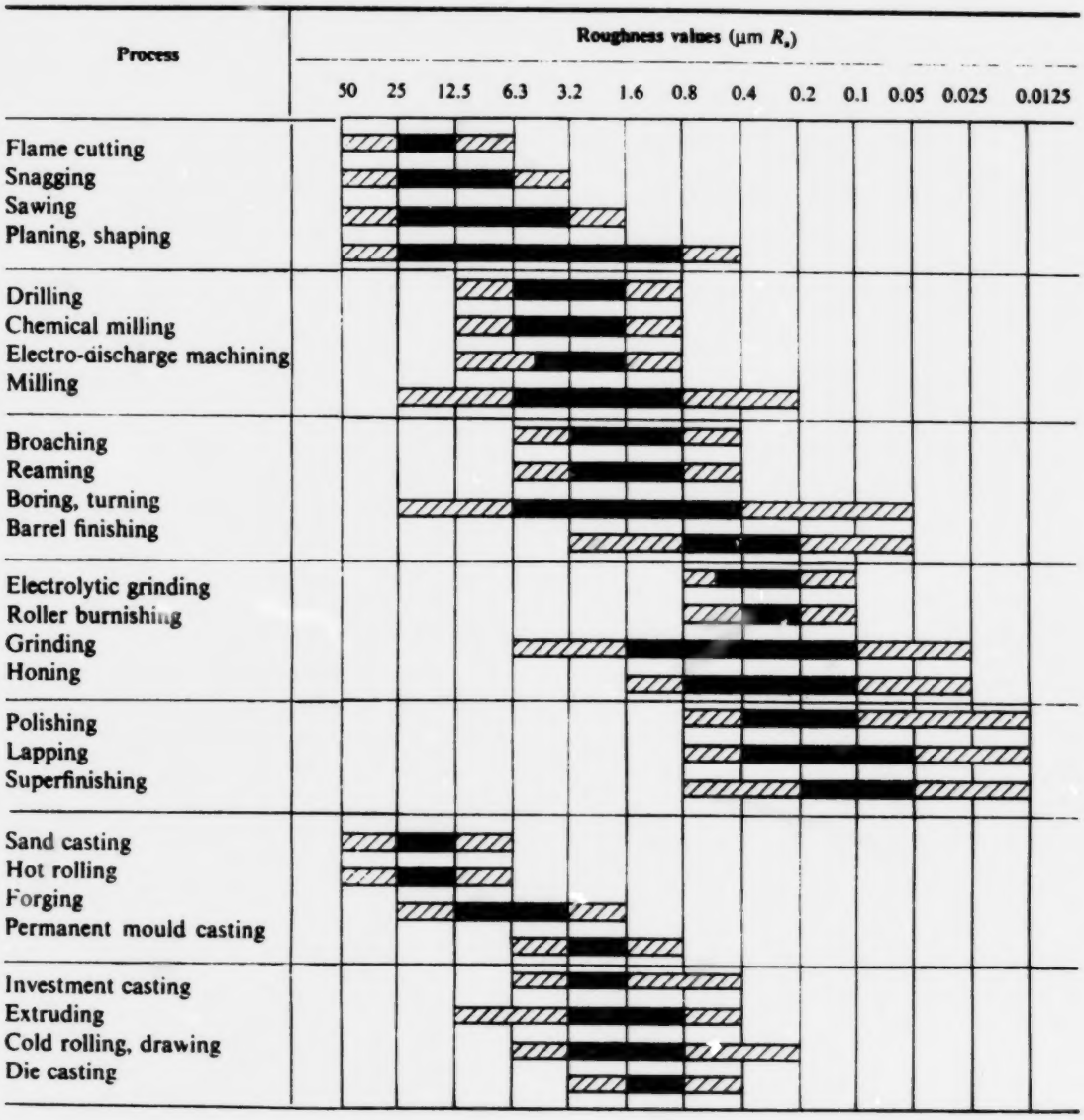


FIG.18 INFLUENCE OF DIFFERENT SURFACE FINISHES ON THE COEFFICIENT OF FRICTION [10]

TABLE 1 SURFACE ROUGHNESS VALUES PRODUCED BY COMMON PRODUCTION PROCESSES AND MATERIAL [45]

Key: ■ average application ▨ less frequent application



NOTE. The ranges shown above are typical of the processes listed. Higher or lower values may be obtained under special conditions.

TABLE 2 EXTRACT FROM ASTM A 480-STANDARD SPECIFICATION FOR GENERAL REQUIREMENTS FOR FLAT-ROLLED STAINLESS AND HEAT-RESISTING STEEL PLATE, SHEET AND STRIP [46]

7. Finish for Sheet

7.1 The types of finish available on sheet products are:

7.1.1 *No. 1 Finish*—Hot-rolled, annealed, and descaled.

7.1.2 *No. 2D Finish*—Cold-rolled, dull finish.

7.1.3 *No. 2B Finish*—Cold-rolled, bright finish.

7.1.3.1 *Bright Annealed Finish*—A bright cold-rolled finish retained by final annealing in a controlled atmosphere furnace.

7.1.4 *No. 3 Finish*—Intermediate polished finish, one or both sides.

7.1.5 *No. 4 Finish*—General purpose polished finish, one or both sides.

7.1.6 *No. 6 Finish*—Dull satin finish, Tampico brushed, one or both sides.

7.1.7 *No. 7 Finish*—High luster finish.

7.1.8 *No. 8 Finish*—Mirror finish.

NOTE 2—*Explanation of Finish*:

No. 1—Produced on hand sheet mills by hot rolling to specified thicknesses followed by annealing and descaling. Generally used in industrial applications, such as for heat and corrosion resistance, where smoothness of finish is not of particular importance.

No. 2D—Produced on either hand sheet mills or continuous mills by cold rolling to the specified thickness, annealing, and descaling. The dull finish may result from the descaling or pickling operation or may be developed by a final light cold-rolled pass on dull rolls. The dull finish is favorable for retention of lubricants on the surface in deep drawing operations. This finish is generally used in forming deep-drawn articles which may be polished after fabrication.

No. 2B—Commonly produced the same as 2D, except that the annealed and descaled sheet receives a final light cold-rolled pass on polished rolls. This is a general purpose cold-rolled finish. It is commonly used for all but exceptionally difficult deep drawing applications. This finish is more readily polished than No. 1 or No. 2D Finish.

Bright Annealed Finish is a bright cold-rolled highly reflective finish retained by final annealing in a controlled atmosphere furnace. The purpose of the atmosphere is to prevent scaling or oxidation during annealing. The atmosphere is usually comprised of either dry hydrogen or a mixture of dry hydrogen and dry nitrogen (sometimes known as dissociated ammonia).

No. 3—For use as a finish-polished surface or as a semifinished-polished surface when it is required to receive subsequent finishing operations following fabrication. Where sheet or articles made from it will not be subjected to additional finishing or polishing operations, No. 4 finish is recommended.

No. 4—Widely used for restaurant equipment, kitchen equipment, store fronts, dairy equipment, etc. Following initial grinding with coarser abrasives, sheets are generally finished last with abrasives approximately 120 to 150 mesh.

No. 6—Has a lower reflectivity than No. 4 finish. It is produced by Tampico brushing No. 4 finish sheets in a medium of abrasive and oil. It is used for architectural applications and ornamentation where high luster is undesirable; it is also used effectively to contrast with brighter finishes.

No. 7—Has a high degree of reflectivity. It is produced by buffing a finely ground surface, but the grit lines are not removed. It is chiefly used for architectural or ornamental purposes.

No. 8—The most reflective finish that is commonly produced. It is obtained by polishing with successively finer abrasives and buffing extensively with very fine buffing rouges. The surface is essentially free of grit lines from preliminary grinding operations. This finish is most widely used for press plates, as well as for small mirrors and reflectors.

The effect of surface roughness of the metallic sliding plate in a TFE bearing with lubrication requires further investigation.

3.10 TYPE OF TFE

Virgin (unfilled) TFE resin is widely used to provide the vertical load carrying sliding surface in a bearing, while filled TFE is commonly used on the guidebars providing lateral restraint. Virgin TFE resin has a lower coefficient of friction than filled TFE as can be seen in Fig.19, which shows the variation of friction coefficient with pressure for both virgin TFE and TFE containing 25% glass fibers. However, the addition of fillers to TFE can significantly improve its resistance to wear and creep [8,11,28].

Unfilled TFE in the form of a woven fabric is used on the vertical load carrying surface in Lubrite bearings manufactured in the United States of America. The TFE woven fiber is pressed onto the geometrically-grooved surface of a metal substrate to form the sliding surface as shown in Fig. 20. Manufacturer's literature [6] indicates that this fabric has a bearing strength approximately 30 times that of virgin TFE resin and that this high bearing strength, together with high shear resistance, virtually eliminates creep under sustained load, However, a bearing pressure of only 24 MPa is used for design in conformity with AASHTO Specifications [25]. A maximum working stress of 52 MPa is allowed on woven TFE fiber on the rotational surface of spherical bearings by a State of Oregon Specification for bearings [47].

Some tests should be carried out on this woven fiber in respect to its application for bridge bearings.

3.11 ATTACHMENT OF TFE TO BACKING PLATE

The Ontario Specification for bearings [19] requires the TFE element to be fully bonded and recessed in a rigid backing material. Bonding is intended to ensure that the TFE element will not be displaced from the recess during the life of the bearing. A case where an unbonded TFE element squeezed out under cyclical movements of a bearing in service has been reported [4]. Bonding will complicate replacement of the TFE in a bearing in service. To avoid bonding, Mageba in Switzerland lowers the temperature of the TFE disc prior to inserting it in the recess, thereby ensuring a tight fit when the TFE returns to the same temperature as the backing plate.

It has been reported [11] that, for a thin TFE sliding pad, 2.5 mm in thickness, the quality of the bonding will affect the coefficient of friction as well as the durability of the pad. A bond containing an excessive amount of voids or bonding material tends to produce a rippled surface on the pad. This irregular sliding surface tends to increase the initial coefficient of friction. However, for the 4.5 mm to 6.0 mm thick TFE normally used with a recess, the presence of excessive voids or bonding material will not be as significant as in a thinner pad. Consequently, it is unlikely that the bonding process will have an influence on the coefficient of friction.

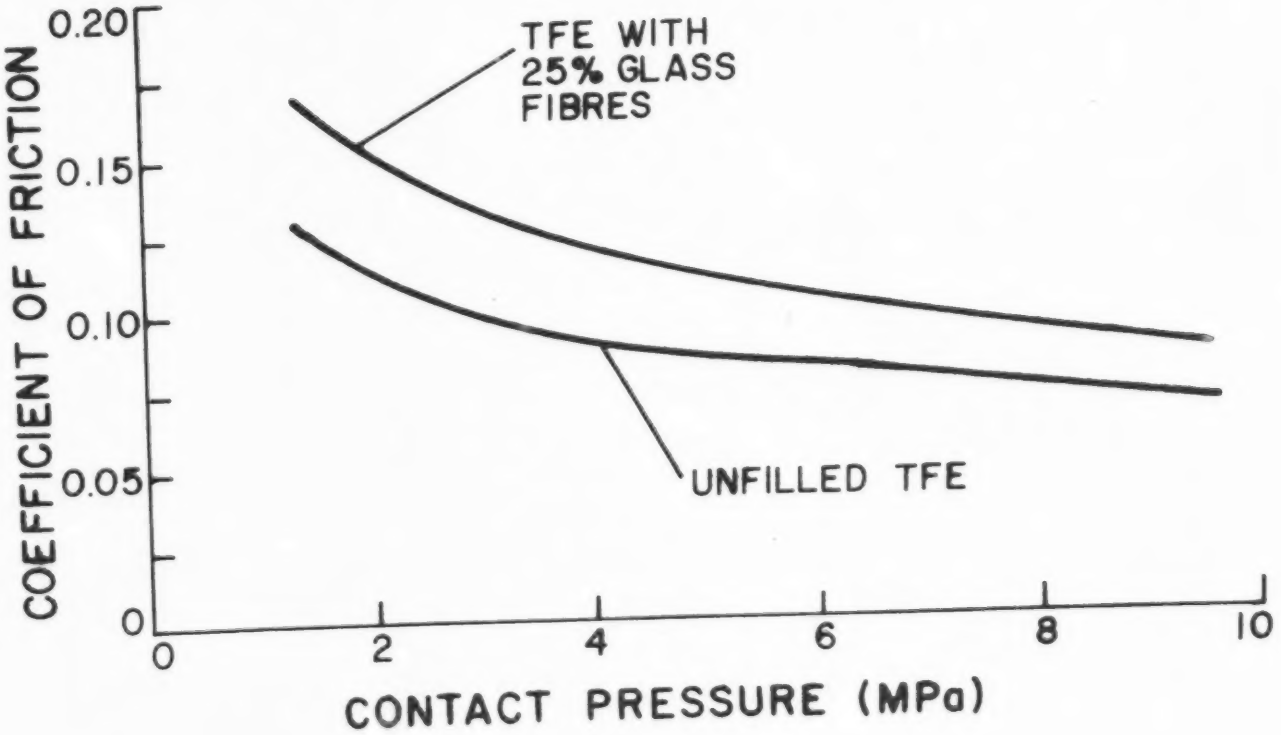


FIG.19 EFFECT OF FILLER ON THE COEFFICIENT OF FRICTION OF TFE [28]

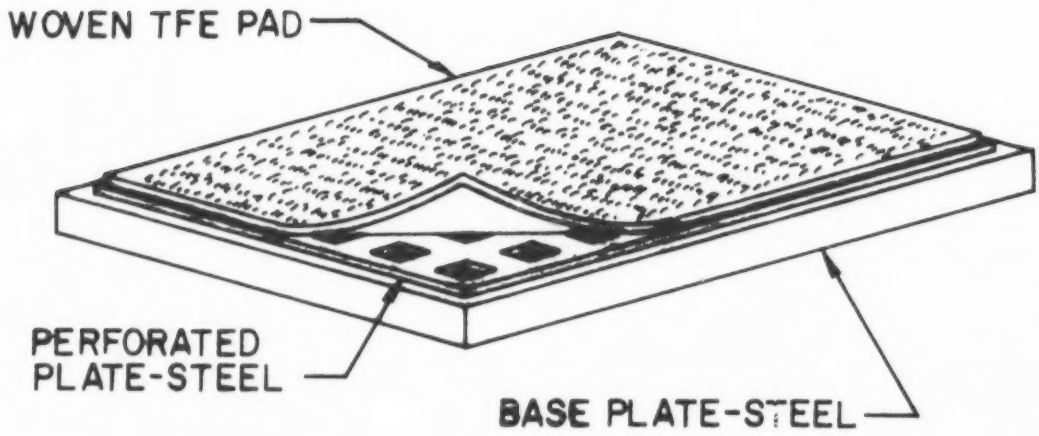


FIG.20 LUBRITE BEARING WITH WOVEN TFE PAD [5]

3.12 SURFACE CONTAMINATION

Virgin TFE is comparatively soft and is capable of absorbing light contamination without any apparent effect on the coefficient of friction and bearing life [15,30,48,49]. On the other hand, it has been reported [15,28] that heavy contamination with materials such as dry cement dust or sand particles substantially increases the coefficient of friction, especially during the initial period of movement. An increase in the static friction coefficient of approximately 10 times has been reported [15] for contamination by dry cement dust. However, light and heavy contamination have not been defined quantitatively. Care should be exercised in the installation of bridge bearings to avoid exposure and contamination of the sliding surfaces in order to obtain satisfactory performance.

Lubrication of the sliding surface tends to accentuate the problem of surface contamination since particles tend to be held to the surface [8]. An increased coefficient of friction resulting from contamination of the lubricant has been reported [18]. Figure 21 shows a distinct increase in friction above a slide path of 5 km when as-rolled instead of polished stainless steel is used. In these tests a strong discolouration of the lubricant occurred due to small particles being released from the as-rolled finish of the stainless steel. Therefore mechanically finished stainless steel surfaces are recommended. This observation suggests that TFE is not transfered to the stainless steel plate when lubrication is used.

In West Germany, airborne contamination of the lubricant in service is prevented by provision of skirts or bellows on the bearing as shown in Fig.22. Some deterioration is still likely to occur in the long term due to the presence of minute particles of dust and debris [8]. However, it is virtually impossible for surfaces affected in this way to be damaged to an extent that would create conditions at the interface leading to a change in the friction. Effectively, the sliding surface would simply become less polished.

The performance of a TFE sliding bearing should be satisfactory provided precautions are taken to protect the TFE surface from contamination during assembly, shipping, installation and in-service. However, it is suggested that tests with known surface contamination should be undertaken.

3.13 LENGTH OF THE TRAVEL PATH

The knowledge of the dependency of the coefficient of sliding friction for TFE upon the total accumulated amount of movement is relatively new. The coefficient of friction increases with total slide path [18,22,33,50]. Figure 23 shows the variation of the frictional coefficient at different temperatures with total accumulated amount of movement up to 25 km. The relationship between the coefficient of friction and the total sliding length may be explained using TFE transfer theory.

With the exception of the initial higher frictional properties (if unlubricated), during the early cycles, a transfer of TFE onto the mating surface takes place. These thin-layered deposits indicate the beginning of abrasion and deleterious effect on the both slide surfaces, resulting

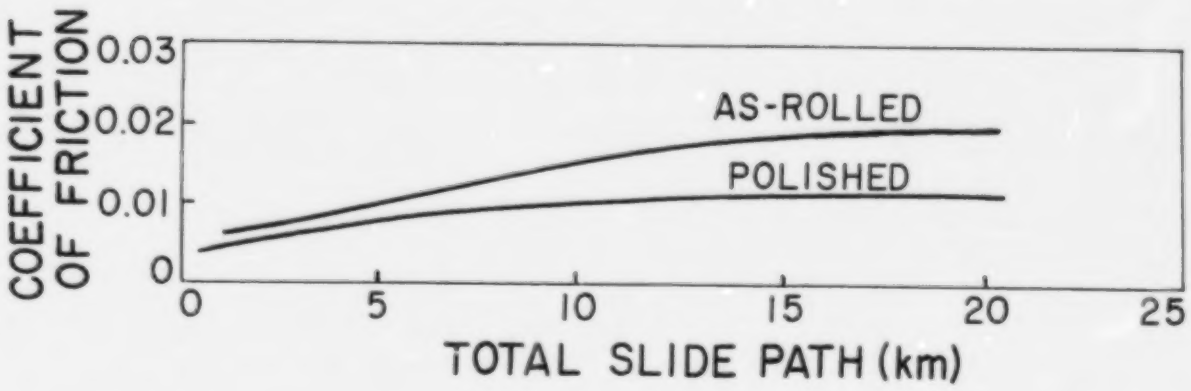


FIG.21 EFFECT OF FINISH ON STAINLESS STEEL PLATE ON THE COEFFICIENT OF FRICTION [18]

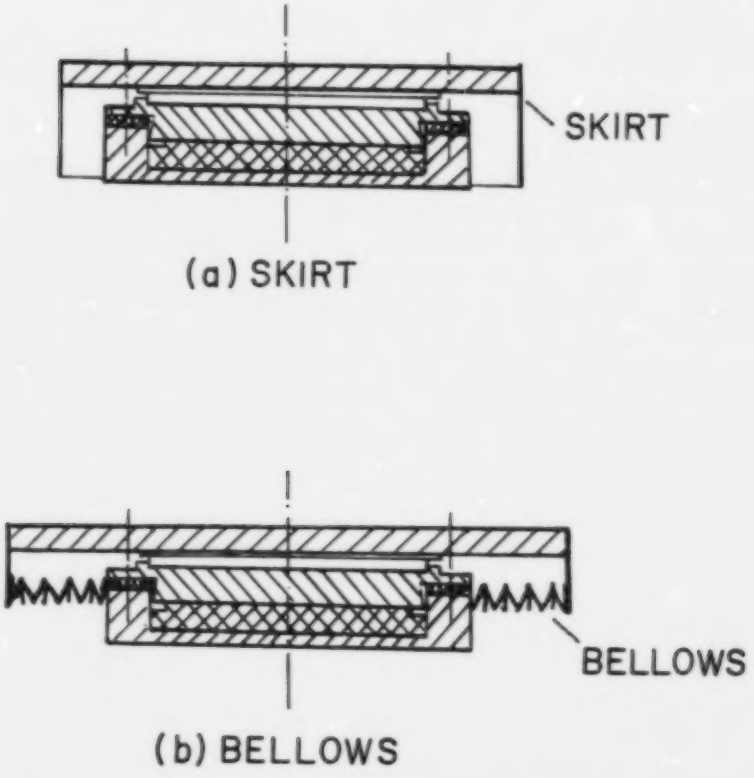


FIG.22 PROTECTION OF SLIDING SURFACES [18]

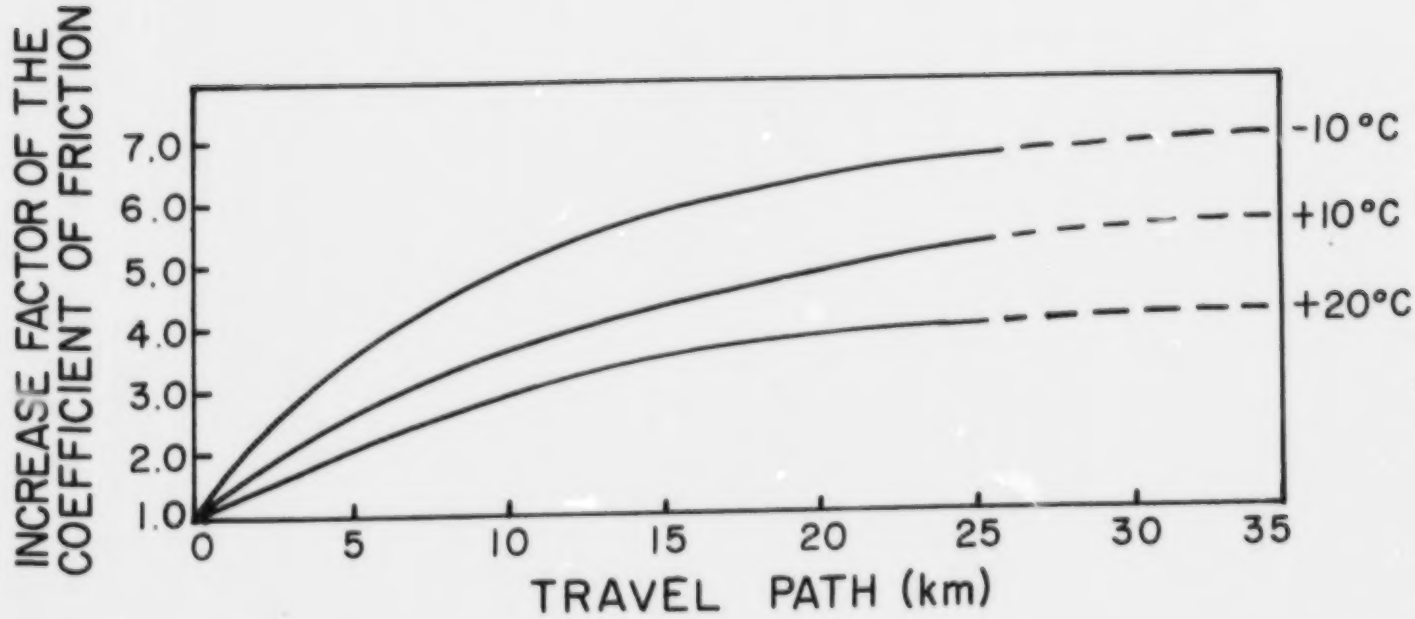


FIG.23 INFLUENCE OF TRAVEL PATH AND TEMPERATURE ON THE COEFFICIENT OF FRICTION [50]

in an increase in friction [22]. With continuous running, after a certain length of sliding path, a constant level of friction is reached, indicating equilibrium between the spreading and pressing out of existing abraded material and the transfer of new particles of TFE. However, it has also been reported [15] that, in a test at room temperature with a pressure of 24 MPa, a sliding speed of 2.5 mm/sec, a stroke of 50 mm and a surface roughness of 0.05 μm R_a stainless steel plate, the coefficient of friction decreased with length of slide path.

The total length of travel path for a bearing in service is so far basically unknown. Recently, tests to measure horizontal bridge movements have been conducted on a number of bridges in the Federal Republic of Germany [37,51]. Preliminary data from two tests on comparable steel and concrete bridges show that traffic induced displacements may be up to 10 times larger for the steel bridge than for the concrete bridge and that, for the steel bridge, temperature induced displacement may be twice that in the concrete bridge. Traffic induced movements ranging from 145 m per year to 5000 m per year [50,51,52] have been reported for steel bridges. A total accumulated amount of movement due to street car traffic of 4570 m per year has been reported [50] for a prestressed concrete bridge.

Researchers at RWTH Aachen believe that up to 50% of the bridge movement is taken up by elastic deformation in a pot bearing with a corresponding reduction in travel path on the TFE. This belief must be examined from tests on full scale bearings.

Total movement of a bridge in service can only be established by site measurements. Further long term field monitoring of such movements is necessary.

3.14 LOAD AND TRAVEL HISTORY

When the load on the TFE is sustained for a few hours, prior to sliding (referred to as dwell), an increase in the coefficient of friction, especially the initial one, is observed [26]. However, a short term interruption, for instance up to 20 minutes, in the sliding movement has no practical effect [35]. Tests [10] reveal that the static coefficient of friction for an unlubricated virgin TFE sample under 24 MPa pressure increased rapidly up to about 24 hours dwell of load and then began levelling off. A maximum friction value was reached after a dwell of about 120 hours. However, it has also been reported [22] that the preloading effect is exhibited only by a lubricated TFE resin sliding surface. AASHTO [25] requires a 12 hour dwell of load prior to carring out a friction test on TFE. Figure 24 shows the variation in the coefficient of friction for a woven fabric TFE due to dwell of load after completion of specific number of cycles. It can be seen that a dwell period results in a friction peak on restarting, although this is below the initial friction value.

The fact that the static friction increases with the dwell of load for new materials seems anomalous to the fact that static friction reduces with increase of pressure, since one would expect time and pressure to work in the same direction. It appears that some factor other than the adjustment of the TFE crystal under load may account for the variation in friction with dwell of load [10].

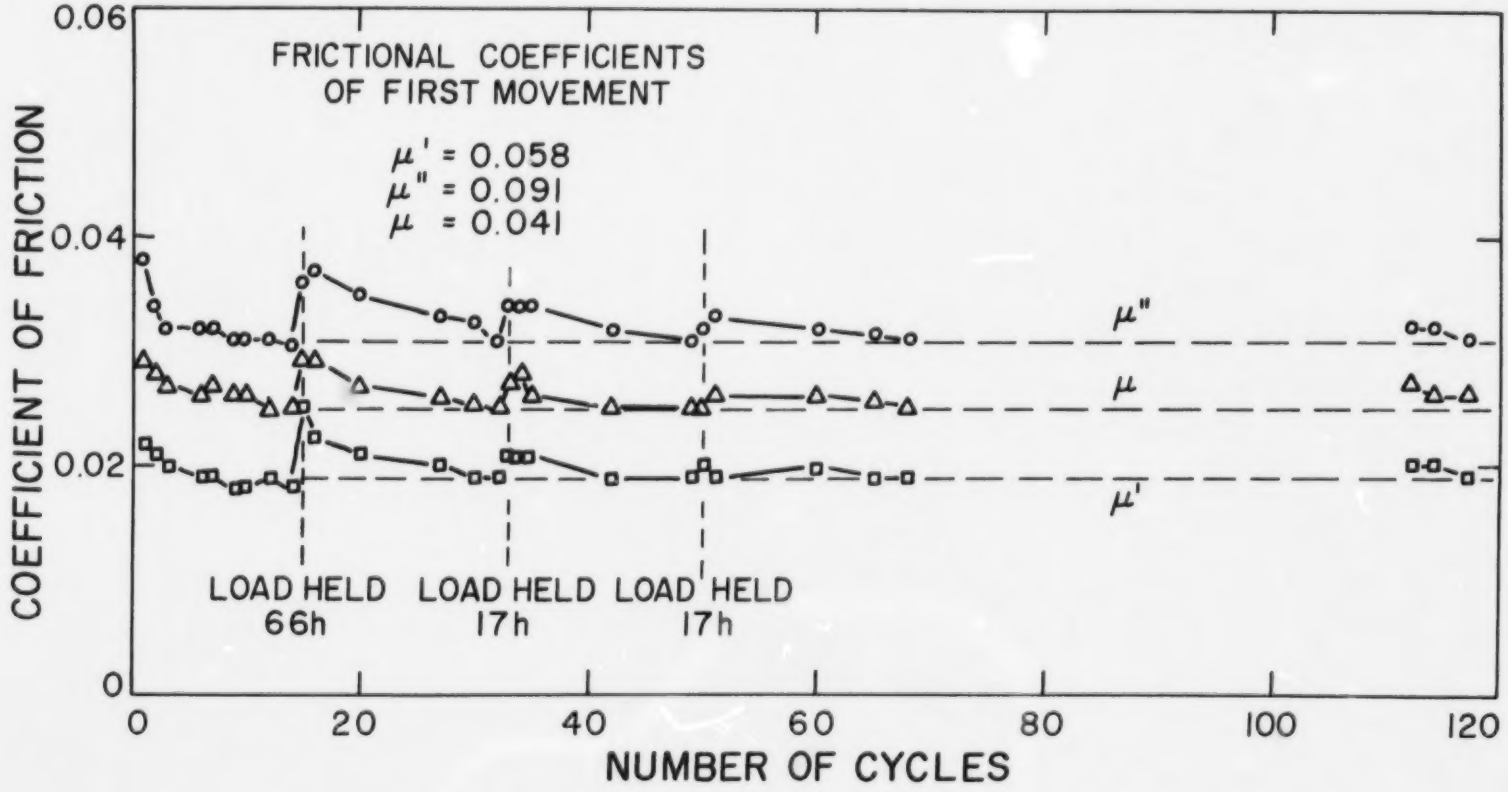


FIG.24 INFLUENCE OF DWELL PERIOD ON THE COEFFICIENT OF FRICTION FOR WOVEN TFE [9]

Horizontal movements in bridge bearings are due to three different causes, namely traffic load, daily temperature change and seasonal temperature change. Movement due to traffic is always superimposed on the diurnal and annual cycles of movement. Extreme low or high temperature in a long span structure may move the TFE contact area onto a little used part of the sliding plate. This will lead to an increase in the coefficient of friction. This effect does not appear to have been investigated, and therefore further studies are required in this area.

3.15 SPECIMEN SIZE

The measured coefficient of friction of TFE has been reported to be lower for smaller specimen size [26]. However, some studies [8,53] indicate that the effect of variation in area of the specimen is minimal as shown in Fig.25. This has been confirmed by tests carried out by Glacier on TFE discs of 75 mm and 150 mm diameter. Discs having a diameter of 75 mm are used commonly in testing TFE in Europe and appear to be the accepted standard.

3.16 WEAR

Wear of the TFE may have an influence on the operational life of a bearing. It may result in a reduction in the thickness of the TFE with subsequent binding of the sliding plate on the plate confining the TFE. Further, the coefficient of friction may increase to an unacceptably high level.

It is normally assumed that wear of TFE can be predicted from the following equation [54]

$$h = KPVt \tag{3.8}$$

where h is the reduction in thickness, K is the wear factor, P is the contact pressure, V is the sliding velocity and t is the time under load. The wear factor depends on whether the application is operating in the mild or severe wear regimes. The mild wear regime (low K value) is characterised by surface layer wear, while the severe wear regime (high K value) involves actual shearing away of the TFE. These two types of behaviour have been observed in the tests carried out at varying sliding speeds and temperature as discussed previously.

It has been suggested [54] that two distinct values of the wear factor exist, one for the mild wear regime and the other for the severe wear regime, with an abrupt transition occuring at the "PV limit". This limit is reached when the surface temperature of the TFE exceeds a critical value. A value of $K = 8.7x10^{-6} mm^2 s/N$ hr has been suggested [17] for a metallic plate with surface roughness of 0.05 μm R_a sliding at the rate of 2.54 mm/sec and a contact pressure of 24 MPa.

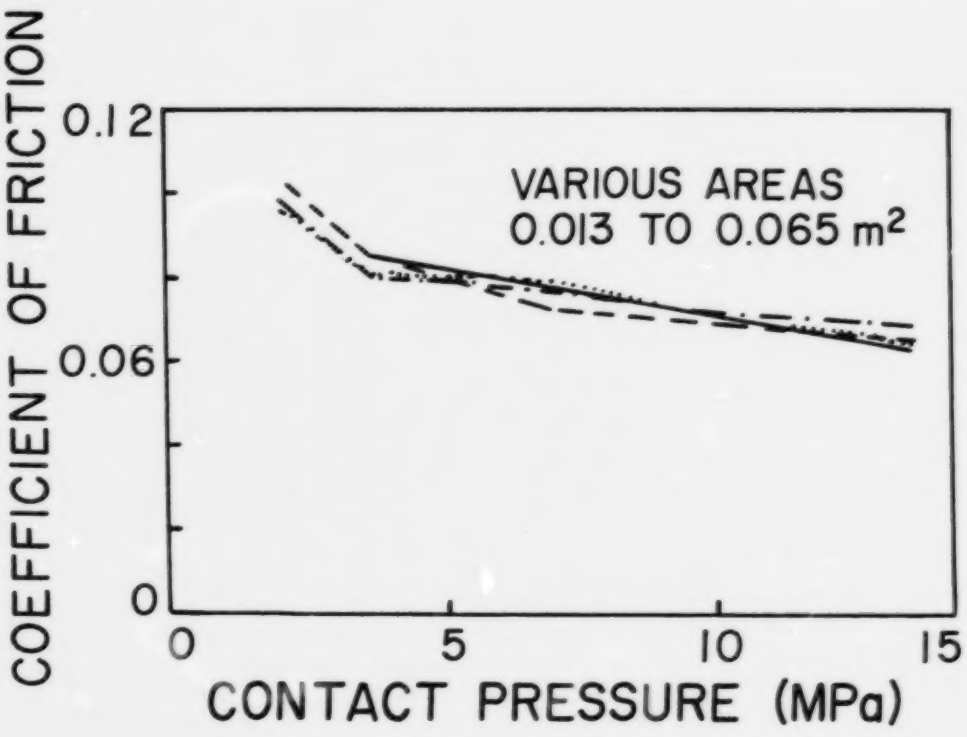


FIG.25 INFLUENCE OF CONTACT AREA ON THE COEFFICIENT OF FRICTION [53]

Wear in a bearing is usually expressed in terms of a wear rate which is defined as the reducion in thickness of the TFE per unit distance of travel of the metallic sliding plate contacting the TFE. From Eqn.(3.8), the wear rate, W, can be expressed as

$$W = h/Vt = KP \tag{3.9}$$

Test data [55], shown in Fig. 26, indicate that the wear rate is dependent on sliding velocity and contact pressure. Other data from this test series indicate that the wear rate is directly proportional to the surface roughness of the sliding plate. Consequently, for Eqn.(3.9) to be valid, the wear factor (K) must be dependent on sliding velocity, contact pressure and surface roughness. Figure 26 also shows the variation in wear rate with the commonly used parameter, namely the product of the contact pressure (P) and the sliding speed (V), for an unlubricated test condition [55]. It can be seen that the wear rate increases significantly for values of PV in excess of 10 MPa mm/sec.

It has been reported [17] that friction and wear are affected in a similar mnanner by variation in most of the related parameters. Consequently, lubrication may be expected to reduce the wear factor. However, the addition of fillers, which tend to increase friction, will reduce wear of the TFE significantly [11,12].

Figure 26(a) indicates that the rate of increase in the wear rate increases with sliding velocity. Consequently, for a particular travel path length, a large number of short faster sliding movements from traffic will be more critical on the operational life of a bearing than a smaller number of long slower movements from thermal effects. It has been suggested [18] that, on very "live" bridges, the TFE may have to be replaced within a period of 5 years because of wear.

Predicion of wear on a TFE surface subjected to a particular contact pressure requires knowledge of the wear factor and total travel path. Little information is available in these items. Consequently a test program on TFE bearings should address these two items.

3.17 TESTING PROGRAM

A number of parameters which influence the frictional properties of TFE sliding surfaces have been identified. Some of these are not well documented and in some cases contradictory effects have been reported. These parameters are contact pressure, temperature, speed of travel, surface contamination, load and travel history, travel path, roughness of the metallic sliding plate and eccentric loading. It is proposed that further studies are required on the influences of these parameters on the frictional properties of TFE sliding surfaces. A suitable laboratory testing program is being developed.

Four test setups, which have been used in previous investigations [8,22,53,56] of the coefficient of frction of TFE, are shown in Figs.27 and 28. In three of these test setups a metallic plate slides between a TFE sheet and a roller bearing (Figs.27(a),(b) and 28(a)). A test

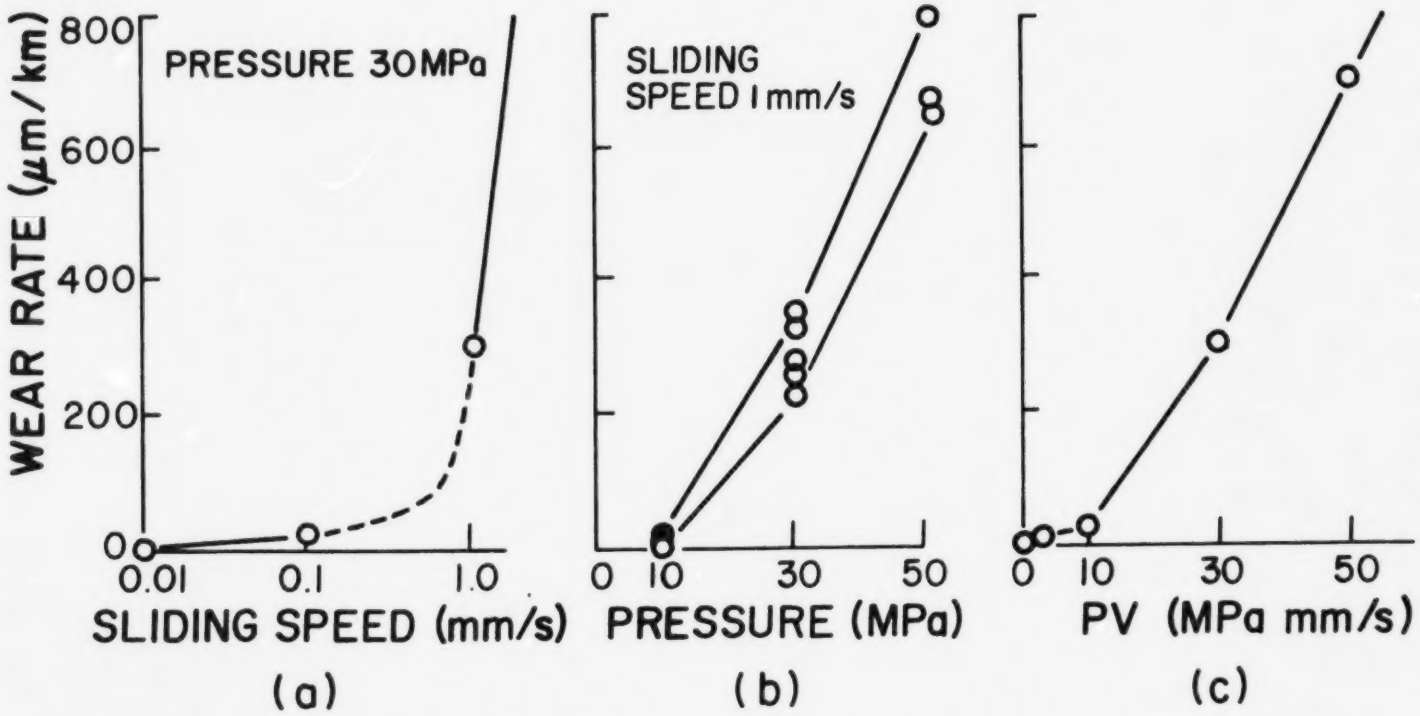
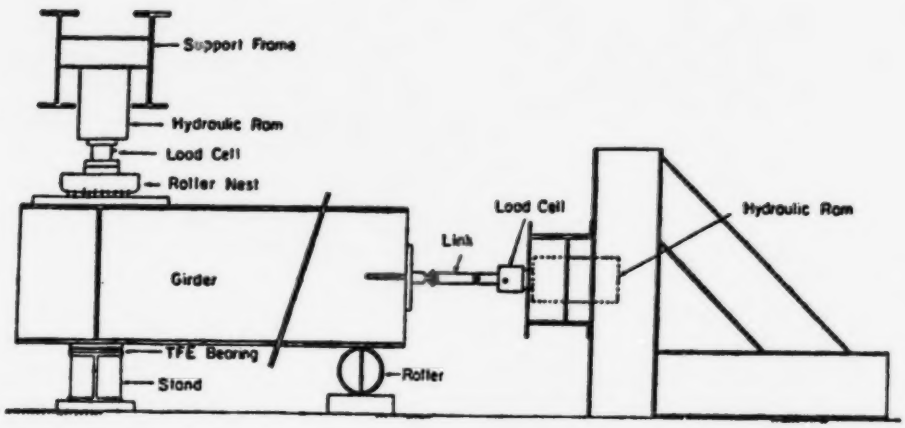
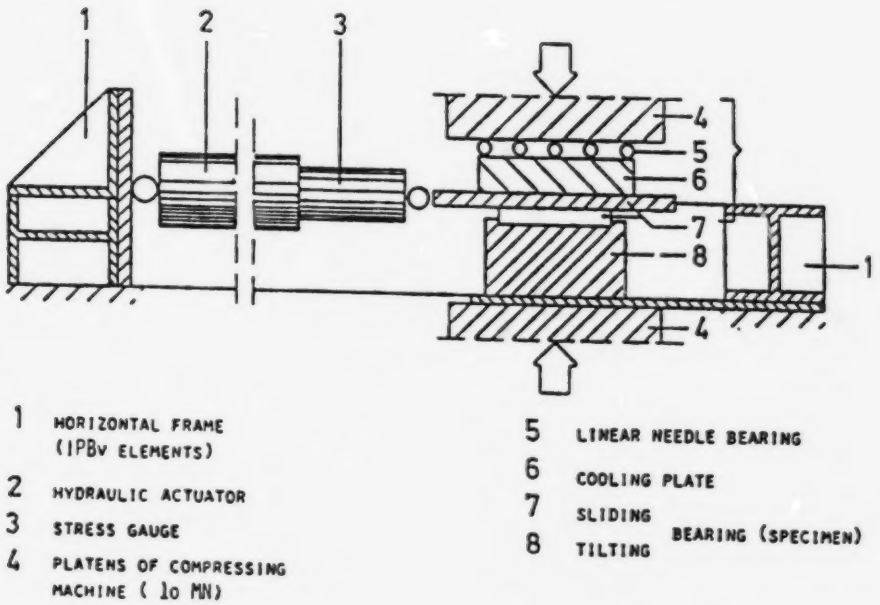


FIG.26 VARIATION OF WEAR RATE WITH SLIDING SPEED, PRESSURE AND *PV* [55]

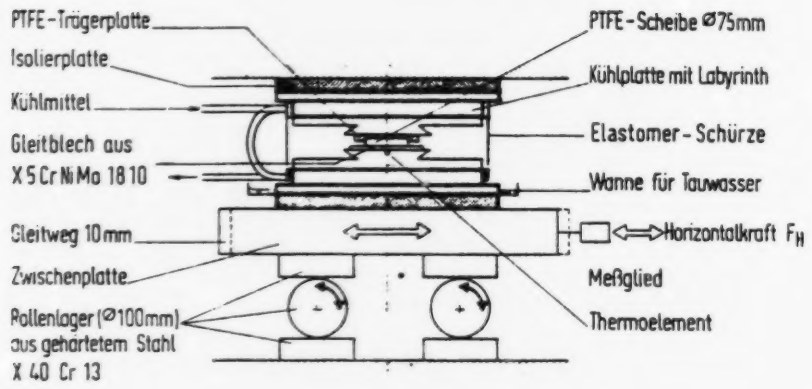


(a) UNIVERSITY OF OKLAHOMA [53]

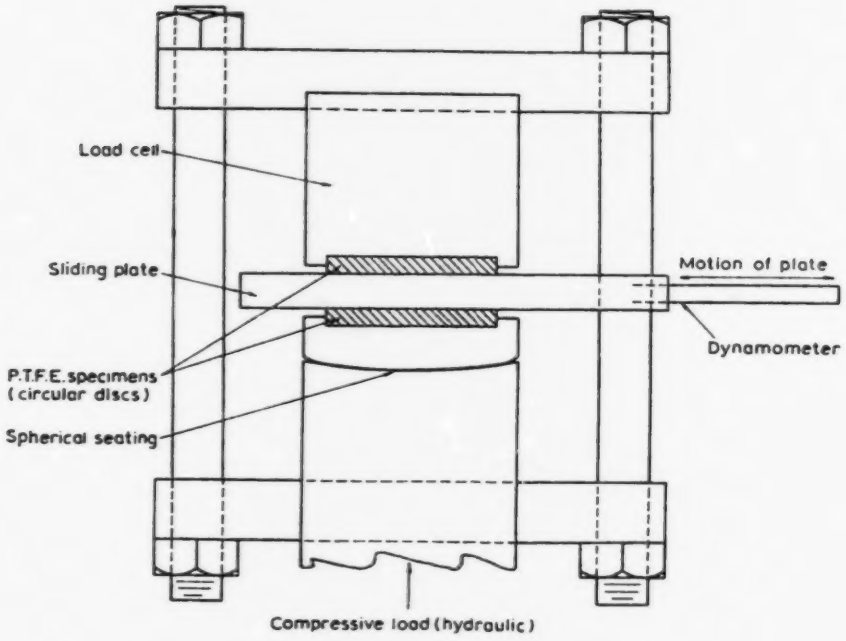


(b) RWTH AACHEN [56]

FIG.27 TEST RIGS USED AT THE UNIVERSITY OF OKLAHOMA AND AT RWTH AACHEN



(a) MPA STUTTGART [22]



(b) TRRL [8]

FIG.28 TEST RIGS USED AT MPA STUTTGART AND AT TRRL

rig in which two TFE samples are compressed on opposite faces of a sliding plate is shown in Fig.28(b). The disadvantage of the test setup in Fig.28(b) is that only an average value of the coefficient of friction for the two TFE surfaces can be obtained. The design of these rigs will be considered in the development of a suitable test rig for the proposed testing program.

3.18 SUMMARY

The coefficient of friction is the prime parameter in the design of a TFE sliding bridge bearing since it dictates the magnitude of the force transmitted from the superstructure to the substructure. The coefficient of friction at the metallic/TFE interface in a bearing is influenced significantly by lubrication, contact pressure, temperature, speed of travel and length of the travel path, as well as to a lesser degree by roughness of the metallic surface, type of TFE, surface contamination, and load and travel history. Effects of some of the parameters are not clearly documented in the literature. In fact contradictory statements exist as to the effects of some of the parameters on the frictional properties, particularly for lubricated surfaces. These parameters are pressure, temperature, speed of travel, surface contamination, load and travel history, and roughness for a lubricated surface. Deformation due to creep of the TFE, specimen size and method of attachment of the TFE to the backing plate will not affect the coefficient of friction. The effect of eccentricity of loading on the coefficient of fricion has not been studied but this effect leads to increased wear of the TFE. Wear may be significant in bridges with pronounced response to vehicular traffic. Further investigation needs to be carried out in a number of areas in order to have a better understanding of the behaviour of TFE bridge bearings in practice.

4. CONCLUDING REMARKS

A comprehensive search of the literature relating to TFE sliding bearings produced 126 relevant publications. Many of these describe research and development work being carried out in West Germany. Only 31 relate directly to TFE bearings as used in North America.

The coefficient of friction is the prime parameter in the design of TFE sliding bearings since it dictates the magnitude of the force transmitted from the superstructure to the substructure. A search of the literature revealed that the coefficient of friction is influenced by 14 identifiable parameters. The influences of some of these parameters are not well documented. Such parameters are contact pressure, temperature, speed of travel, surface contamination, load and travel history, travel path, roughness of the sliding plate and eccentricity of load. It is recommended that the influences of these parameters be investigated in a comprehensive test program.

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APPENDIX A

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APPENDIX B

LOCATIONS AND PERSONNEL VISITED IN EUROPE

1. Institut fur Bautechnik, Berlin, W. Germany
   - Dr. Ing. H. Eggert, Dr. Ing. Voss and Dipl. Ing. Buche

2. Institut fur Bauforschung, Aachen, W. Germany
   - Dr. Ing. H.R. Sasse, Dipl. Ing. B. Schwamborn, Dipl. Phys. A. Boue and Dipl. Ing. M. Breitbach

3. Mageba SA, Bulach, Switzerland
   - W. Koster and R. Huber

4. Maunsell & Partners, London, England
   - R. Bristow

5. Glacier Metal Company Ltd., Ilminster, England
   - M. Baigent and J. Baker

6. Department of the Environment, Coleraine, N. Ireland
   - J. Kirkpatrick